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The feasibility of applying vibration monitoring techniques to high volume multistation transfer machines.

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THE FEASIBILITY OF APPLYING VIBRATION MONITORING TECHNIQUES TO HIGH VOLUME MULTISTATION TRANSFER MACHINES

by

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ABSTRACT

A study was undertaken to determine the feasibility of applying vibration monitoring techniques to high volume multistation transfer machines.

Recent published literature on machinery health monitoring is reviewed with special emphasis on vibration monitoring. A complete bibliography of 255 references is appended, together with summary chart, in which the subject is classified by topics.

A field study was undertaken to determine the feasibility of applying vibration monitoring techniques to high volume, multistation transfer machines installed in one of the leading automotive engine plants. An accelerometer and a tape recorder were used to obtain the vibration data. It was shown that repeatable vibration measurements were possible under "in plant" conditions and that future trends in both the overall and spectral acceleration levels were readily apparent. Furthermore, for one particular machining station, prediction of bearing failure was documented.

As a result of the successful "in plant" manual vibration monitoring, a series of controlled bearing failure tests were performed in order to determine the most suitable vibration analysis technique for identifying specific types of failure. The results of defects which were induced to

the outer race, the inner race, one ball and subsequently as a combination of all three, on a bearing of a typical single spindle machining station, were carefully studied. The spindle was operating at approximately 1680 rpm without metal cutting. Accelerometers were again used in the vibration measurements. Generally, it was concluded that each induced defect (except a ball defect) could be identified from the frequency analysis as well as time domain analysis. The reasons for the difficulty in identifying an induced ball defect were also discussed. Furthermore, incorrect bearing installation in the spindle was also identified by frequency analysis.

Thus it was concluded that vibration monitoring could be successfully applied to high volume, multistation transfer machines and that bearing failure could be predicted. Three areas for future research and development are identified: evaluation of bearing condition monitoring techniques, the economics and benefits of using vibration monitoring systems on transfer machines and future vibration monitoring systems.

DEDICATION

This work is dedicated to my wife, JASMIN and our first born who is yet to hear the Noise and Vibrations of this world.

ACKNOWLEDGEMENTS

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TABLE OF CONTENTS

Page

LIST OF FIGURES

Figure		Page

Figure		Page

Figure		Page
5.2b	The Shaft and Bearing Assembly	111
5.3	The Spindle Layout Including All the Measuring Positions	112
5.4	The Schematic of Experimental Set Up	113
5.5	An Induced Outer Race Defect	117
5.6	An Induced Ball Defect	117
5.7	An Induced Inner Race Defect	118
5.8	Induced Multiple Defects	118
5.9a/b	Comparison of Frequency Spectra of Prior to Removal and After Being Removed and Reassembled	124
5.10a/b	Comparison of Frequency Spectra of Incorrectly Installed Bearing and Correctly Installed Bearing	125
5.11a	Acceleration Versus Time for Outer Race Defect (Position 5B)	127
5.11b	Outer Race Defect Spectrum Versus Baseline Spectrum for Position 5B, 0 - 1.6 kHz	127
5.11c	Outer Race Defect Spectrum Versus Baseline Spectrum for Position 5B (Decibel Scale), 0 - 1.6 kHz	128
5.11d	Outer Race Defect Spectrum Versus Baseline Spectrum for Position 5B, 0 - 25.6 kHz	128
5.12a	Acceleration Versus Time for Ball Defect (Position 6B)	132
5.12b	Ball Defect Spectrum Versus Baseline Spectrum for Position 6B, 0 - 1.6 kHz	132
5.12c	Ball Defect Spectrum Versus Baseline Spectrum for Position 6B (Decibel Scale), 0 - 1.6 kHz	133

Figure		Page

LIST OF TABLES

NOMENCLATURE

A/D	analog to digital
ADC	analog to digital converter
AM	amplitude modulation
A_n	the average after n time records
A_{n-1}	the average after (n-1) time records
A Spec	auto spectrum
BW	bandwidth
cm	centimeter
cpm	cycle per minute
d_B	the ball diameter
dB	decibel
D_I	the inner race contact diameter
D_O	the outer race contact diameter
D_P	the pitch diameter
DR	direct
f_A	the ball assembly (fundamental train) frequency
f_B	the ball spin frequency
f_{ht}	the highest frequency of the transition band
f_I	the ball pass frequency of the inner race
f_{in}	the input frequency
f_{max}	the maximum frequency of interest
f_{min}	the minimum frequency of interest

f_O	the ball pass frequency of the outer race
f_R	the rotational shaft frequency
f_S	the frequency of the ADC sampling operation
FFT	fast fourier transform
FM	frequency modulation
g	unit of acceleration (9.81 meter / second ^{2})
g-SE	unit of acceleration of spike energy
Hz	Hertz
in	inch
I_i	the i^{th} time record
I_n	the n^{th} time record
IRD	Indusrial Research and Development Corporation
ISO	International Standards Organization
kHz	kiloHertz
K	the number of samples in a time record
ℓ_B	the linear travel of ball center
ℓ_I	the linear travel of inner race
ℓ_O	the linear travel of ball on the outer race
L.	left
LPF	low pass filter
mil	0.001 inch
ms	millisecond
MAG	magnitude
n	the number of time records
N	the decay constant

N_B	the number of balls
rms	root mean square
rpm	revolution per minute
R.	right
R #	reading number
sec	second
SE	spike energy
STA.	station
T	the total time record length
V_B	the linear velocity of the ball
V_I	the linear velocity of the inner race
#A	the number of averages
β	the angle change of the inner race
ϕ	the contact angle
α_I	the angular travel of inner race
α_O	the angular travel of ball center
Δt	the time interval between digital history values
Δf	the frequency resolution
Σ	the summation

I. INTRODUCTION

Today, achieving higher productivity is the single most important goal of machine tool builders. This, however, requires not only innovative concepts in the part transfer and metal removal processes, but also the integration of monitoring and diagnostic tools that permit early warning of machine component failure. Vibration monitoring is one of the main techniques used to predict and diagnose a wide range of incipient failures in rotating machines. Such capabilities would substantially reduce the problem of unscheduled maintenance, minimize additional damage to the machine, permit advanced planning of changes in production schedules, reduce spare parts inventory and return the machine to operating condition quickly.

Therefore, it is obvious that machine tool builders who wish to succeed in the highly competitive market place of the future, must plan today for the integration of vibration monitoring systems into their machines.

A transfer machine line is a collection of automatic machining stations of all types (see Figure 1.1). The workpiece, such as an engine block, enters the transfer line as a rough casting and leaves it completely machined at a rate of approximately 200 units per hour. The rate of production is very high, thus unscheduled machine downtime

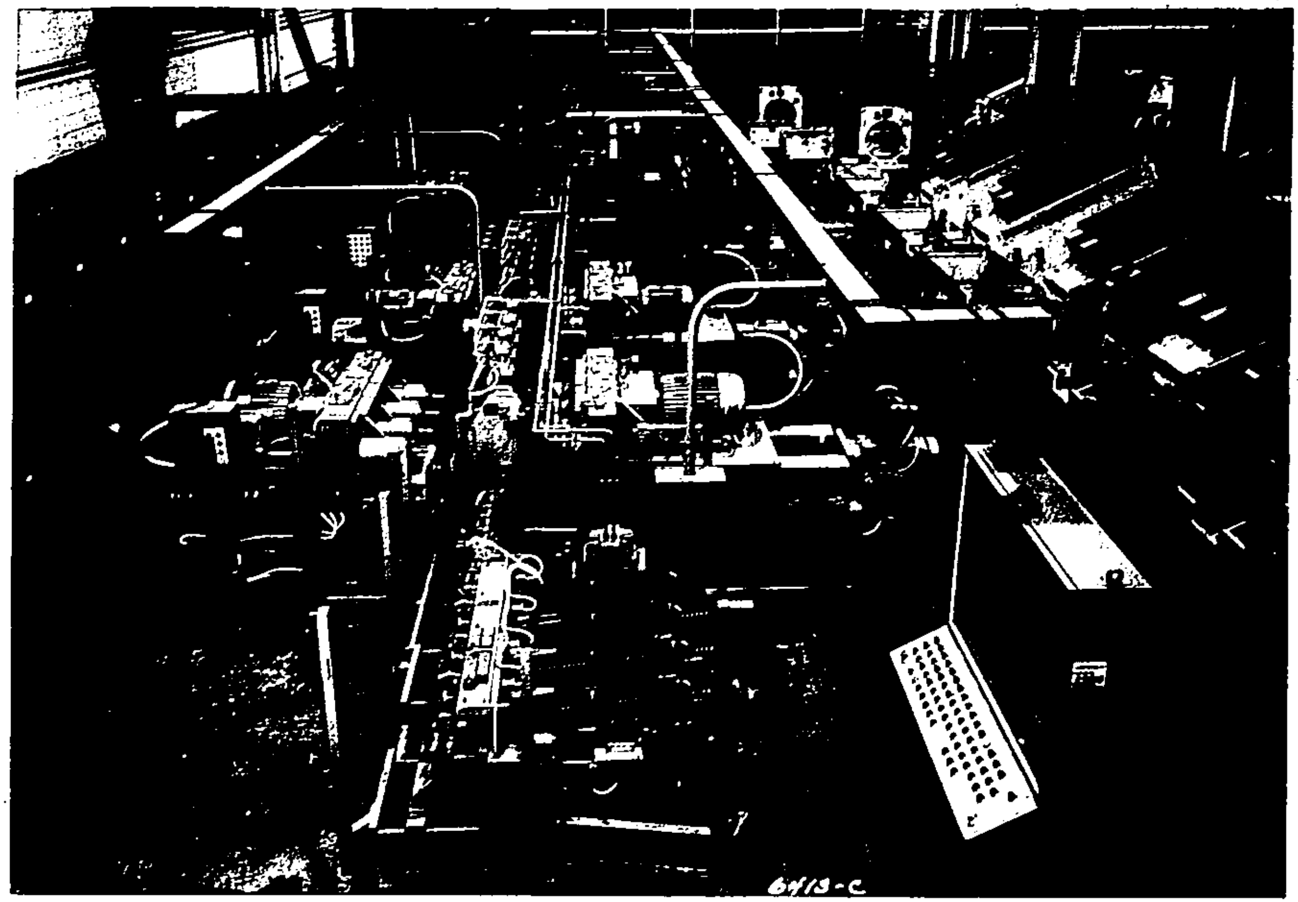


FIGURE 1.1: TYPICAL HIGH VOLUME MULTISTATION TRANSFER MACHINE

can significantly reduce productivity. Furthermore, the failure of one machining station in a transfer line will automatically stall the whole line production. Bearing failures are at present the most common causes of machine downtime. Every bearing has a limited life which is strongly influenced by the method of installation, operating conditions and maintenance received. Thus the reliability, efficiency and safety of the spindles used in the machining station depend on bearings functioning properly.

In view of these considerations the following objectives were set for this thesis:

a. To review the recently published literature on machinery condition monitoring with special emphasis on vibration monitoring techniques.

b. To determine the feasibility of applying vibration monitoring techniques on the high volume, multi-station transfer machines installed in the engine plant of a leading automotive manufacturer.

c. To obtain representative overall acceleration levels as well as frequency spectra from designated machining operations for use as a "baseline" in a future vibration monitoring system.

d. To identify induced bearing defects in a typical single spindle machining station using time domain analysis as well as the frequency domain analysis.

e. To summarize the results and recommend areas of future research and development.

II. LITERATURE SURVEY

The subject of machine health monitoring has long been a widely-documented field in machinery research. In fact, current published literature focuses largely on this subject as an important tool in extensive machine studies. This particular review concentrates on vibration monitoring with special emphasis on bearing and gear failures and automated (computerized) vibration monitoring systems. The topics reviewed include the sources of vibration in rotating machinery, instrumentation, measurement techniques, data processing techniques, applications to various types of machinery and systems, and users' experiences.

Because textbooks offer a very limited amount of practical information on vibration monitoring and analysis of rotating machinery, this review has predominantly made use of periodicals, proceedings and seminar notes. A complete bibliography of 255 references is given in alphabetical order in Appendix A. A summary chart of the full bibliography with classification of technical papers by topics is given in Appendix B. Of the references listed in Appendix A, 155 have been reviewed in detail and are referred directly in this thesis. These entries are embodied in a separate "References" section.

2.1 Maintenance Based on Machinery Condition

For a machine health monitoring program to be successful, it must be integrated into a complete machinery maintenance program which also includes record-keeping, design review, and quality control. The concept of machine health monitoring and diagnosis is not new and its application is gaining considerable attention in the industrial sphere. In fact, recent economic trends and technological advances are leading many companies to take a closer look into the direct and indirect implications of machine monitoring programs which include:

a. the spiralling costs of labor and replacement parts;
b. the energy and production losses when the machine fails to perform at peak efficiency;
c. more powerful and yet lighter weight machinery;
d. the demand for higher reliability and safety; and
e. the development of programmable intruments and the trend towards lower cost of computers.

It may, then, be safe to assume that future trends will emphasize the significant benefits of maintenance based on machinery condition rather than the traditional scheduled or breakdown maintenance strategies. However, for such condition-based maintenance to be possible, it is essential to have sufficient knowledge of machine condition and its rate of change with time so that maintenance can be scheduled in an orderly fashion. This has been shown to reduce unnecessary overhaul of machinery in good condition as well as the cost of maintenance due to unexpected breakdowns. To illustrate this point more concretely, Clark (43) plotted maintenance effort versus time in the graph shown in Figure 2.1.

Neale and woodley (110) presented a paper based on a survey of machine health monitoring in the British industry. The advantages obtained by the application of machine health monitoring to industrial machinery are summarized in Table 2.1. According to the survey, approximately £750 millions (January 1978 value) annually has been estimated as the maximum conceivable savings which could be obtained by applying machine health monitoring across the whole of the British industry. Figure 2.2 illustrates the contributions to this total figure made by the various industrial sectors. The average gross savings was estimated at about one percent of added value output (differences between total sales revenue and cost of material and energy) with a range of 0.5 to 3.0 percent for the various contributions.

Realistic figures on costs and benefits of using machine health monitoring system are rather difficult to obtain. Jackson (84) states that a large petrochemical plant had an annual ten percent improvement (amounting to approximately

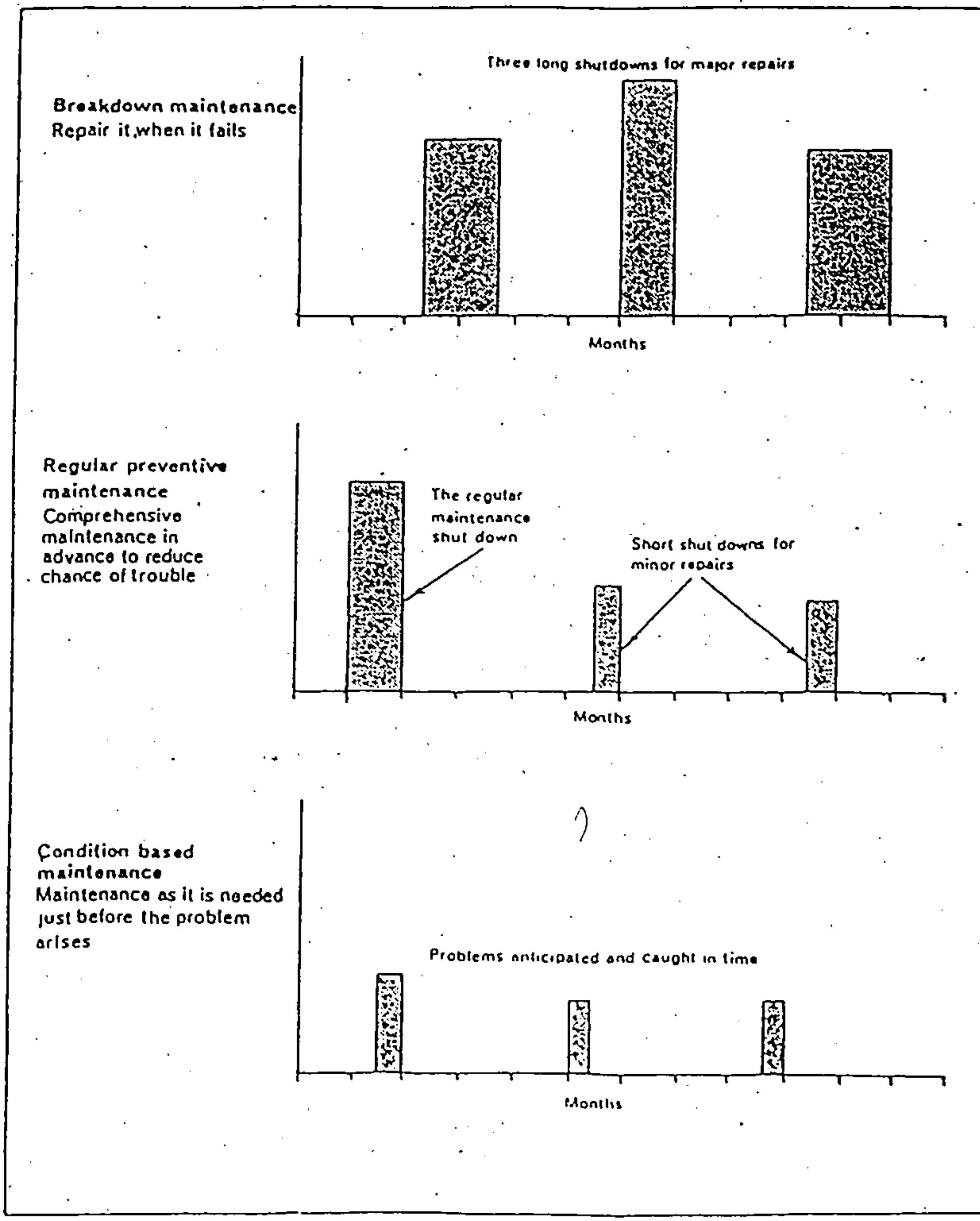


FIGURE 2.1: MAINTENANCE SYSTEMS (43)

<table>
<tr><th colspan="3" rowspan="2">ADVANTAGES OBTAINED</th><th colspan="2">METHODS BY WHICH CONDITION MONITORING GIVES THESE ADVANTAGES</th></tr>
<tr><th>Lead Time</th><th>Better Machine Knowledge</th></tr>
<tr><td>SAFETY</td><td colspan="2">Reduced Injuries and Fatal Accidents to Personnel caused by Machinery.</td><td>Enables plant to be stopped safely when instant shut down is not permissible.</td><td>Machine condition, as indicated by an alarm, is adequate if instant shut down is permitted.</td></tr>
<tr><td rowspan="4">OUTPUT</td><td rowspan="2">Increased Machine Availability</td><td>More Running Time</td><td>Enables machine shut down for maintenance to be related to required production or service, and various consequential losses from unexpected shut downs to be avoided.</td><td>Allows time between planned machine overhauls to be maximised and, if necessary, allows a machine to be nursed through to the next planned overhaul.</td></tr>
<tr><td>Less Maintenance Time</td><td>Enables machine to be shut down without destruction or major damage requiring a long repair time.

Enables the maintenance team to be ready, with spare parts, to start work as soon as machine is shut down.</td><td>Reduces inspection time after shut down and speeds up the start of correct remedial action.</td></tr>
<tr><td colspan="2">Increased Rate of Net Output</td><td></td><td>Allows some types of machine to be run at increased load and/or speed.

Can detect reductions in machine efficiency or increased energy consumption.</td></tr>
<tr><td colspan="2">Improved Quality of Product or Service</td><td>Allows advanced planning to reduce the effect of impending breakdowns on the customer for the product or service, and thereby enhances company reputation.</td><td>Can be used to reduce the amount of product or service produced at sub-standard quality levels.</td></tr>
</table>

TABLE 2.1: THE ADVANTAGES OBTAINED BY THE USE OF CONDITION MONITORING (110)

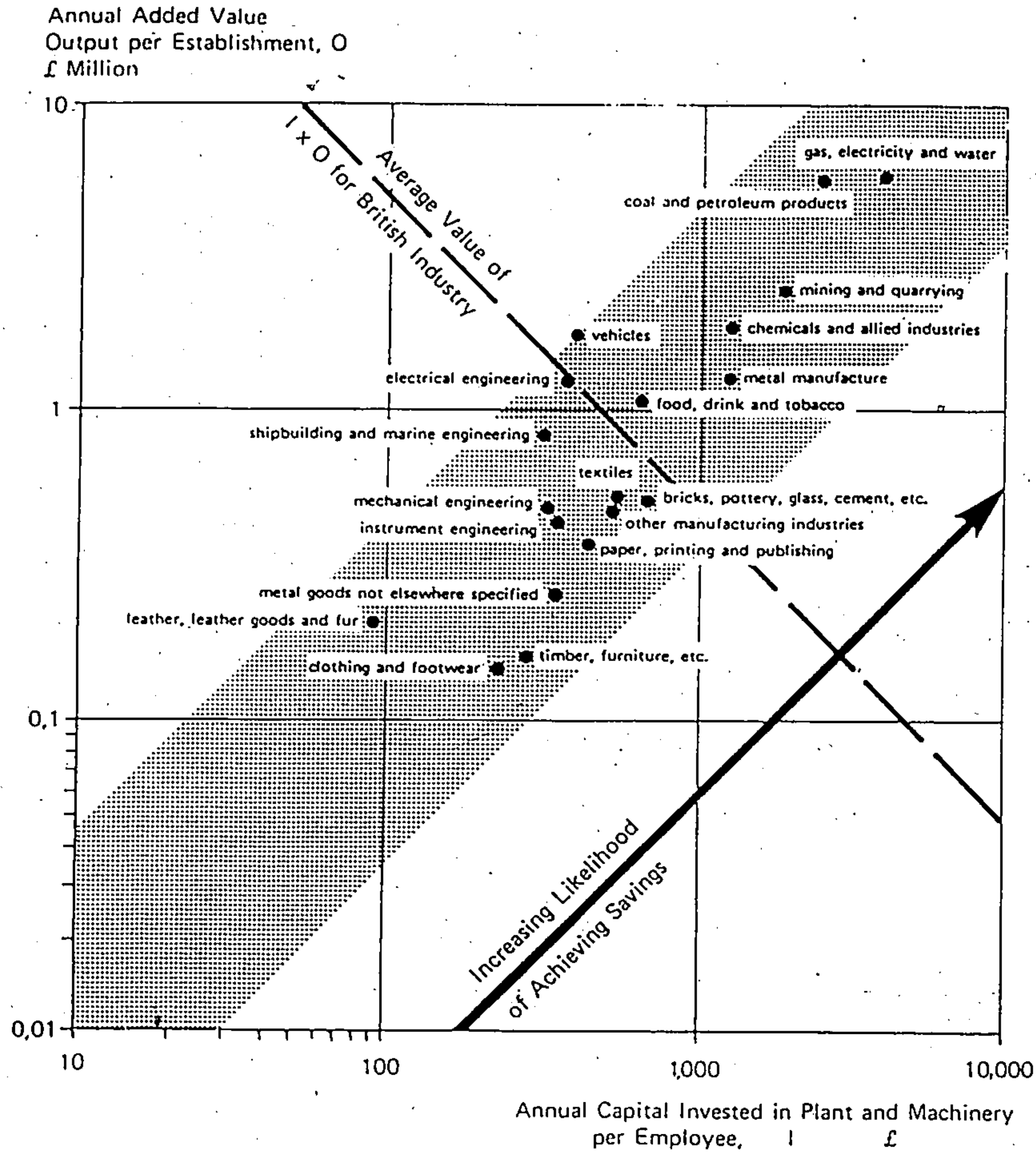


FIGURE 2.2: THE SELECTION OF INDUSTRIAL SECTORS WHICH ARE SUITABLE FOR CONDITION MONITORING [JAN. 1978 VALUES] (110)

$250,000 per year) in the maintenance expenditure during the first five years of implementing a vibration monitoring program. Dodd (49,50) claims a reduction of about thirty percent annually in the maintenance cost for all turbo-machinery units in one major petrochemical plant over a span of seven years. Furthermore, Hudacheck and Dodd (82) found that the vibration monitoring program resulted in the extended overhaul cycles for major machinery from three- up to seven-year intervals.

Apparently, vibration monitoring is being practised in various industrial sectors at different levels of sophistication. This particular topic will be reviewed in more detail in the "Users' Experiences" section.

2.2 Characteristics of Machinery Vibration

Practically all operating machines produce vibration. In the process of performing the job, forces are generated which excite the individual parts of the machine directly or indirectly by affecting the entire machine structure. For as long as the process is constant (or only varying within certain limits), the amount of vibration measured will, likewise, be constant. A deterioration in the machine's running condition almost always produces a corresponding increase in the vibration level. In other words, by monitoring the vibration level, it is possible to obtain information about a machine's over-all condition (4,18,28,95).

Other parameters, such as temperature and oil pressure, also change as the machine condition deteriorates, although at a much later point in the development of a fault than does the vibration level. Moreover, noise measurements can also reveal machine deterioration (6,7,47,52,106,146). However, noise is often more difficult to associate with a particular source of machine failure condition in factories and workshops where many machines are running at the same time in the same area. In any event, since the main sources of faults in industrial machinery have a mechanical origin, it is logical to choose a mechanical phenomenon as the representative parameter of machine condition. Mechanical vibration has proved to be one of the most reliable parameters used for checking machine condition in machine health monitoring (4).

The initial step in implementing a successful machinery vibration monitoring program is the selection of the components to be monitored. This should be based on the vulnerability of either the production or the operational phase to failure of a particular machine (65,105). In addition, the type of service, the availability of spares, and the operating experience should also be considered in the selection of machinery to be monitored and the monitoring system at large.

In designing a machine vibration monitoring system, sufficient knowledge of the sources of machinery vibration is important. Main sources of vibration in transfer lines include the following (8,30,38,140-144):

a. critical speed,

b. natural frequency,

c. structural resonances,

d. unbalance,

e. misalignment,

f. vibrations of individual components including bearings and gears,

g. mechanical looseness and

h. bad drive belts.

It is noteworthy to mention that small changes in process conditions, alignment, or structural stiffness often affect vibration levels of machinery to a large extent. For example, if the bearing assembly belt loosens up, the selective stiffness of the bearing structure or foundation is liable to change and may ultimately alter the critical speed of the shaft.

2.3 Standards and Criteria

Vibration monitoring is a comparative process. Once obtained, vibration data can be compared to baseline data (40), previously obtained from machinery, or to vibration data from identical machinery known to be in good condition. Vibration data can also be compared to established standards. Several common Machinery Vibration Severity Standard Charts are available. Among them are the Rathbone Chart of 1939 (128), the 1964 IRD Chart (40), Blake's 1964 Chart (13), and the PMC/BETA Chart (124). They are illustrated in Figures 2.3 to 2.6. The ISO (International Standard Organization) 2372 classifies the rating of vibration severity of each class of machinery. Eshleman (64) provides further standard charts on specific types of machinery.

All of the above mentioned standard charts are based on a finite sample of machinery and experience and, hence, must be used with discretion. Furthermore, it is worth noting that no absolute values of the severity of machinery vibration signatures exist: the same value could mean satisfactory operation on one type of machine and imminent failure on another (40,78,113).

Downham and Woods (54) developed a technique which permits bearing impedances to be easily obtained. This data can then be used, together with operational vibration levels

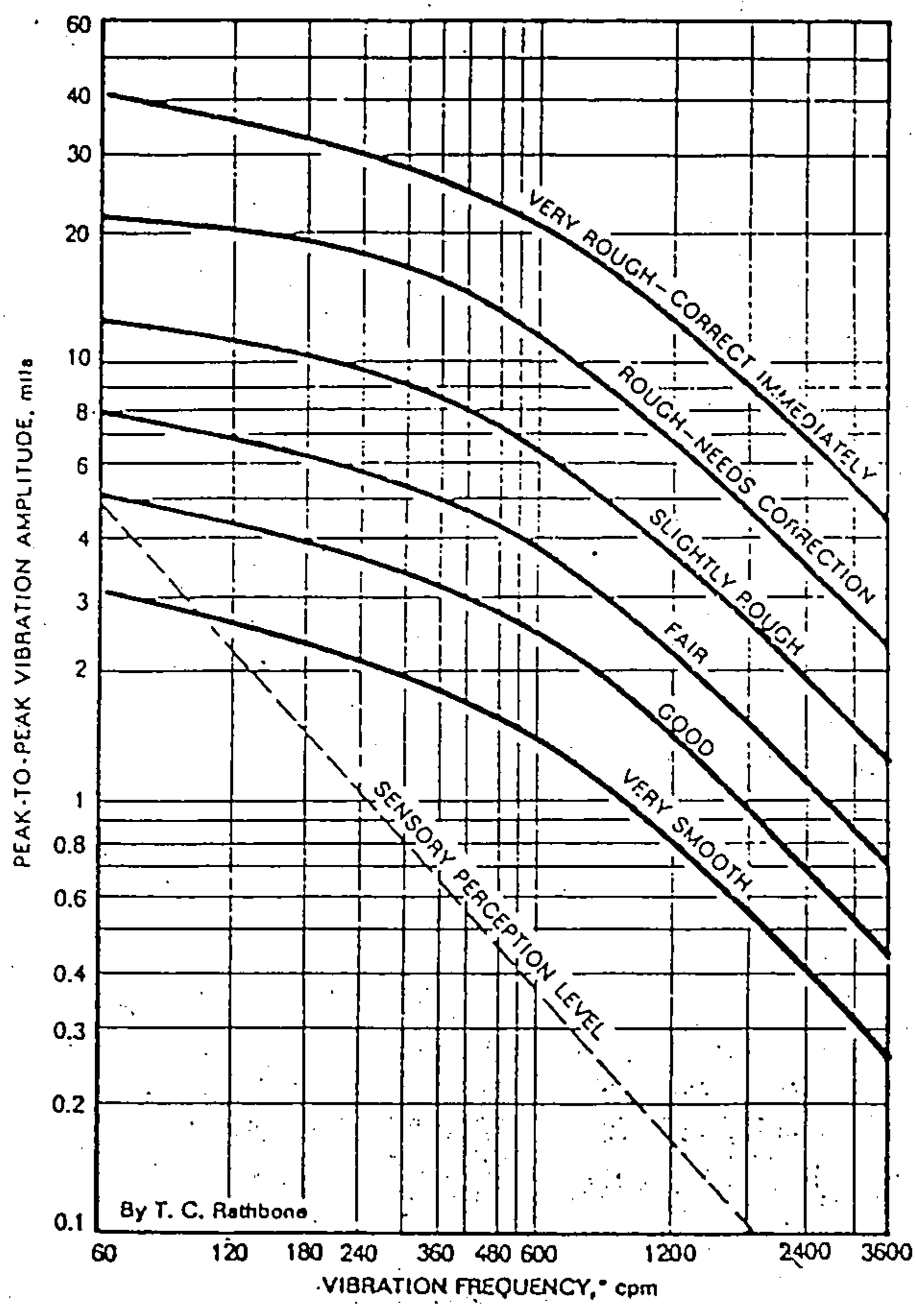


FIGURE 2.3: RATHBONE VIBRATION SEVERITY CHART (128)

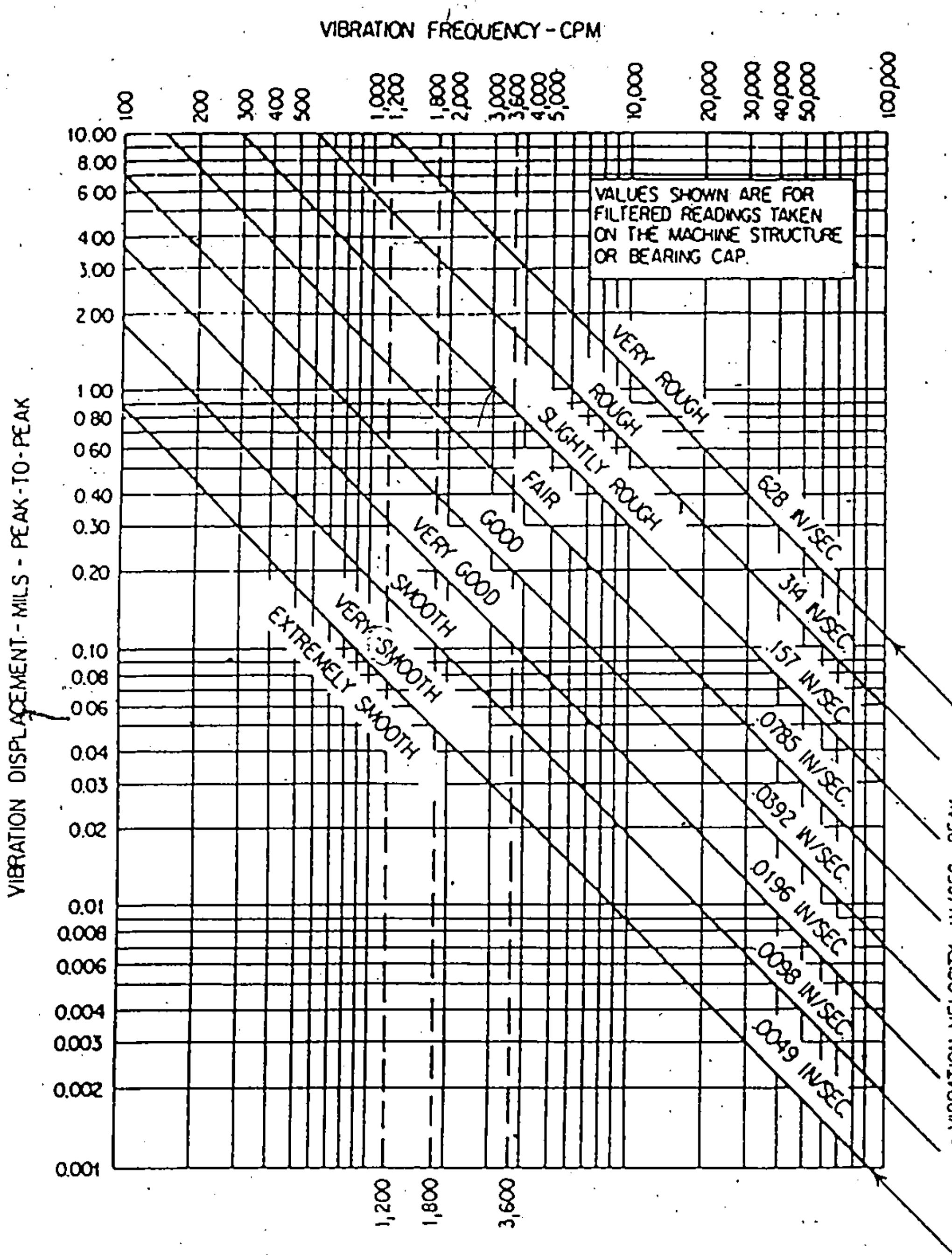
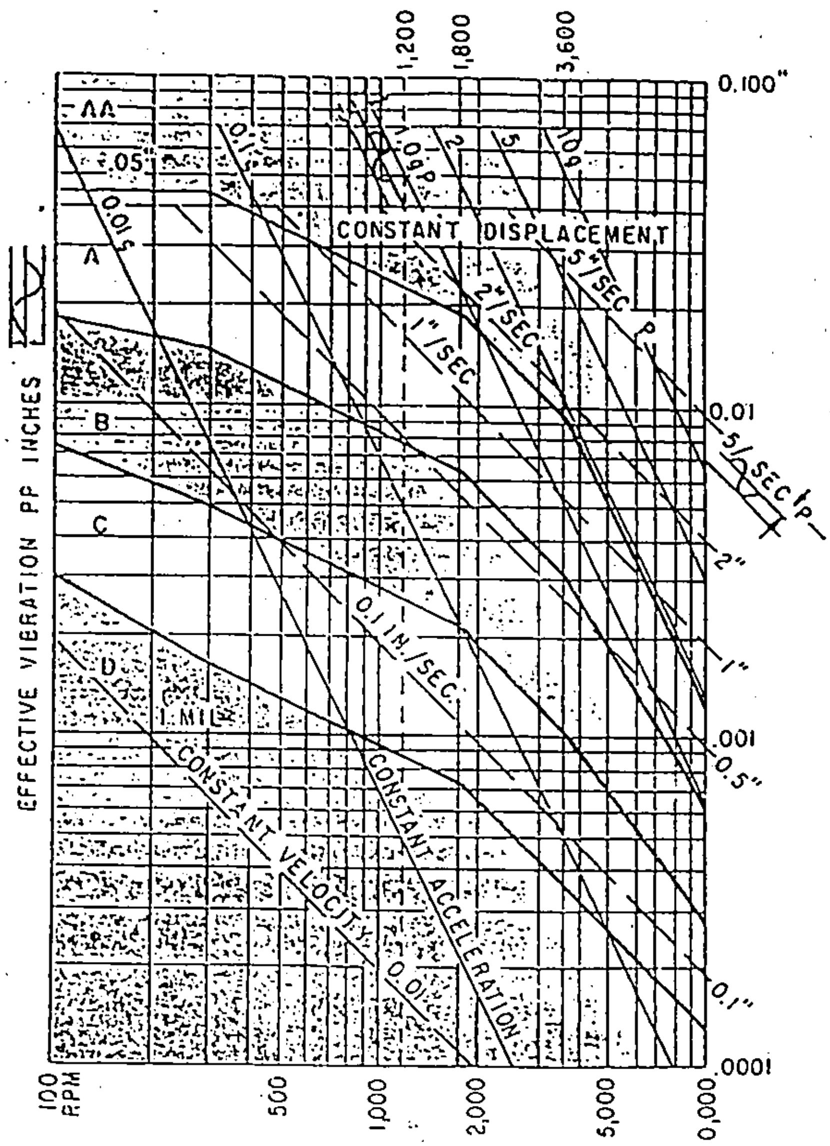


FIGURE 2.4: IRD GENERAL MACHINERY VIBRATION SEVERITY CHART (38)



Explanation of Classes

AA Dangerous. Shut it down now to avoid danger.
A Failure is near. Correct within two days to avoid breakdown.
B Faulty. Correct it within 10 days to save maintenance dollars.
C Minor faults. Correction wastes dollars.
D No faults. Typical new equipment.

This is a guide to aid judgment, not to replace it. Use common sense. Use with care. Take account of all local circumstances. Consider: safety, labor costs, downtime costs.

FIGURE 2.5: VIBRATION SEVERITY CHART BY M.P. BLAKE (13)

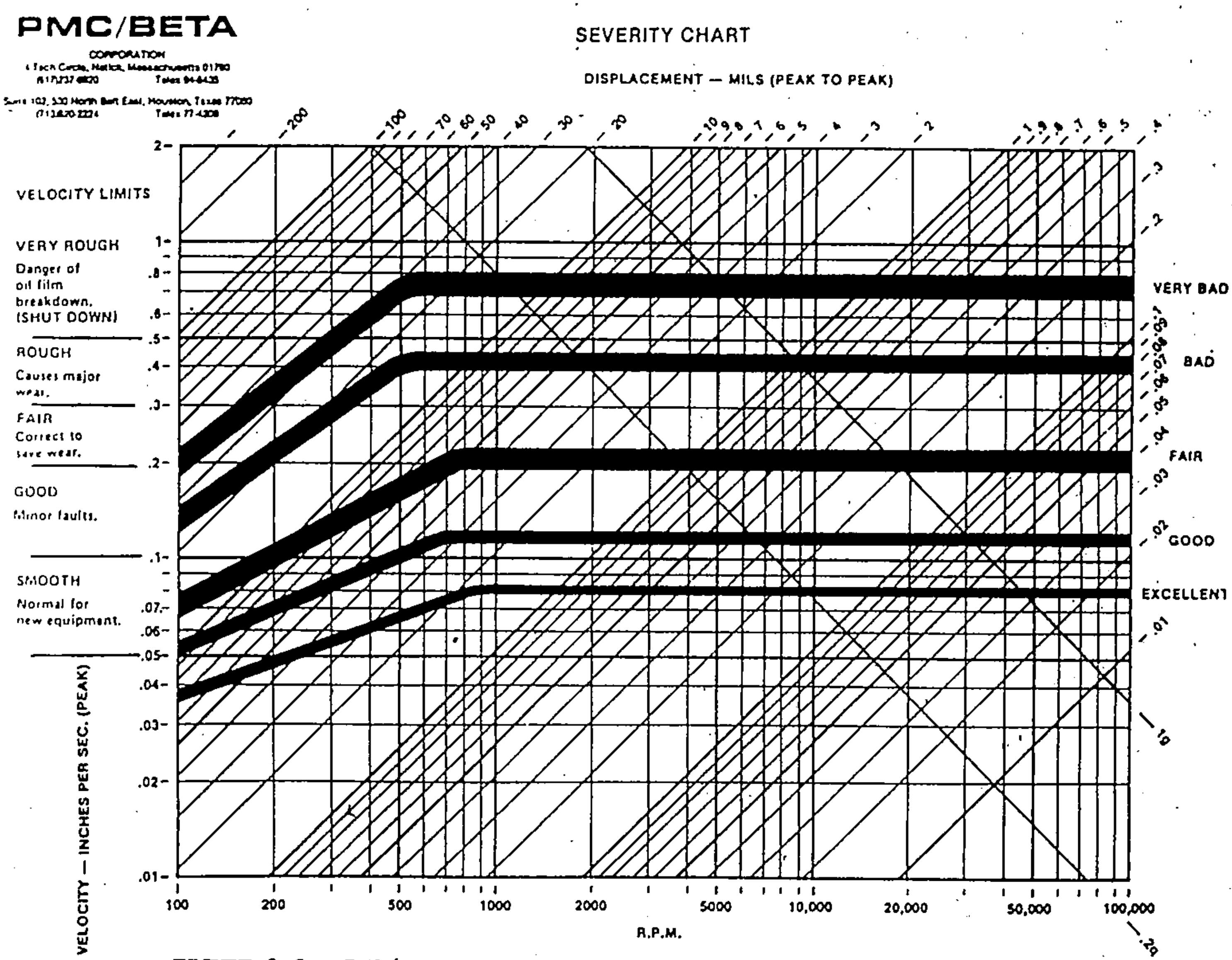


FIGURE 2.6: PMC/BETA CORPORATION VIBRATION SEVERITY CHART (124)

to interpret, in a more realistic manner, general vibration severity charts.

One main objective of vibration monitoring is to predict when the condition of a machine has deteriorated to the point that some type of failure is imminent. However, the significance of an abnormal measurement poses some problems that possibly limit the monitoring process.

It was observed that in some vibration monitoring techniques, random fluctuations that are inherent in the monitored vibration data of machinery are not taken into account (131). Changes up to about 200 percent in vibration level are not uncommon (125). Consideration of the actual random fluctuations in the measured data is, thus, extremely important in assessing the state of a machine since it is possible for a vibration signal to momentarily exceed the safe level. A procedure for determining the onset of failure involves the use of random variations of data in a vibration record to predict both the first passage probability of a vibration signal above a safe level as well as the probability of duration of the vibration measurement exceeding the safe level (131).

Collacott (45,46) has proposed a technique for determining the condition of a machine and predicting its remaining life. The operating life of a machine is divided into two components, namely: infant mortality, in which failures might result from initial defects, and life mortality, in which failures occur due to eventual wear-out. Simple equations are formulated for predicting the time of minimum probability of failure at the end of the infant mortality region and certain failure in the life mortality region. In effect, the whole life of a machine can be, thus, predicted.

Figure 2.7 illustrates the overall vibration level through time for three operating regions of a mechanical system. The objective of machinery vibration monitoring is to determine the point where the rate of change in vibration level has increased. This figure is applicable to machinery baseline signatures described below.

Baseline signatures are simply a carefully obtained set of vibration signatures for a machine in good mechanical condition, operating under standard conditions and in its final installed configuration. A meaningful baseline signature cannot be obtained unless these conditions are satisfactorily achieved (39,40,136). Baseline signatures have been used to uncover many mechanical problems during start-up of new or overhaul equipment (136). This baseline technique for vibration monitoring of machinery has been widely used in many industries (2).

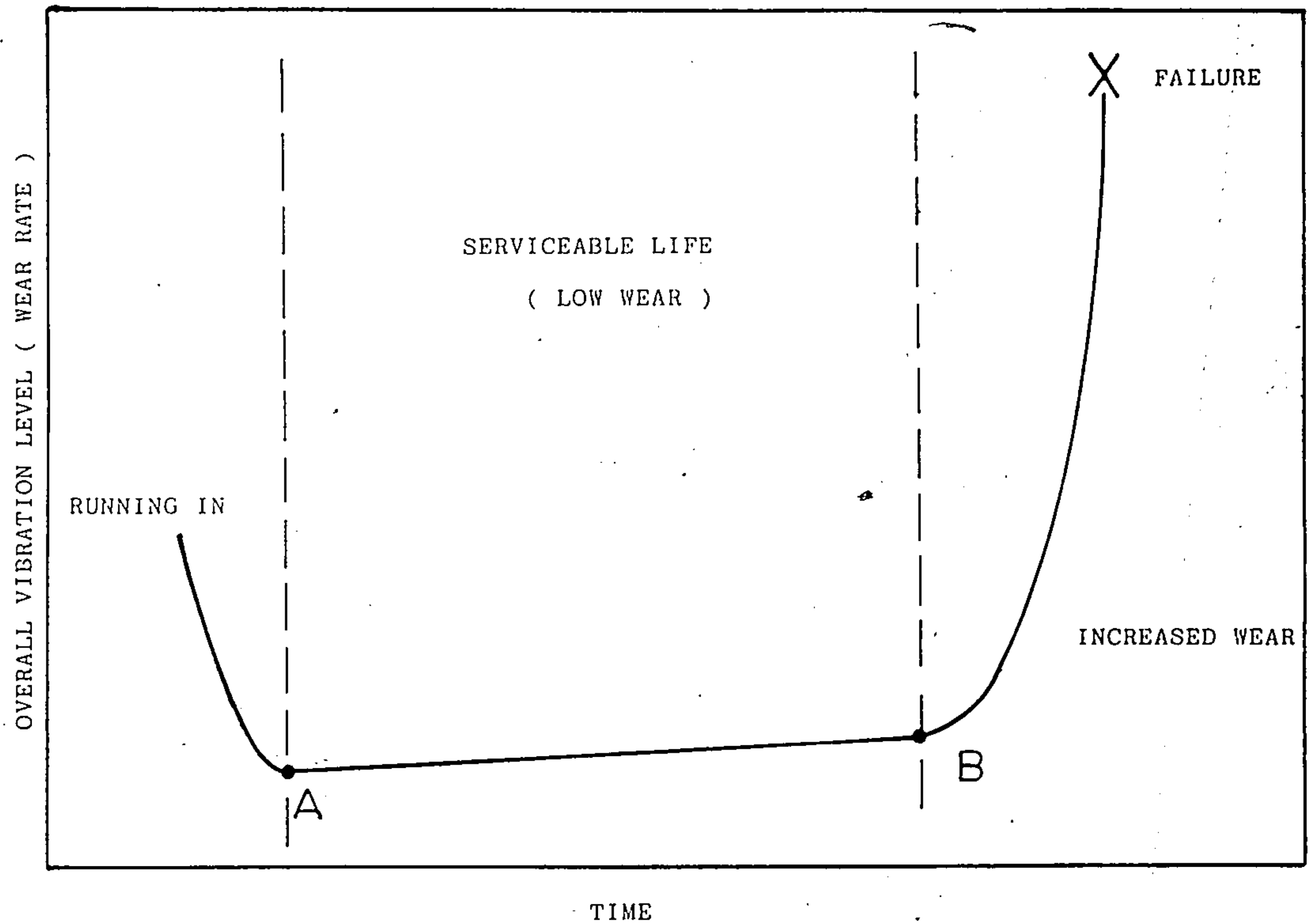


FIGURE 2.7: .VIBRATION LEVEL VERSUS TIME FOR THREE OPERATING REGIONS OF A MECHANICAL SYSTEM

2.4 Measurements and Instruments

2.4.1 Typical Transducers

Three parameters, each having several advantages, are commonly used to describe machinery vibration: displacement, velocity, and acceleration. Recognizing the fact that the analysis of machine health can only be as good as the measurement transducers, choice of a transducer must be made with a specific application in mind. There are three most widely employed transducers: displacement probe, velocity pick-up, and accelerometer. One type of the displacement probe consists of a coil excited by a high frequency oscillator. Varying the distance between the coil and core, in this case the shaft surface, produces a change in inductance and an output voltage proportional to distance. The velocity pick-up consists of a seismically mounted magnetic core surrounded by a coil of wire which is attached to the vibrating surface. Relative motion between the coil and core produces an output voltage proportional to velocity. An accelerometer consists of a seismically mounted mass bearing on a piezoelectric crystal. The crystal produces an output proportional to force which, in turn, is proportional to acceleration at a known mass.

Figures 2.8 to 2.10 are schematic illustrations of the displacement probe, velocity pick-up, and accelerometer, respectively. There are many publications on this topic, as well as a number of manufacturers' brochures (27,68, 73, 84,87,99,104,112,133,150,152) that review the characteristics, applications, advantages and limitations of each of these three most commonly used transducer types.

The choice of which parameter to measure and which transducer is most appropriate depends upon the machine to be monitored. Wear is a function of amplitude and frequency; velocity is the product of these two factors. Thus, velocity tends to identify significant vibration frequencies over quite a broad band. Displacement and acceleration, on the other hand, tend to emphasize the minimum and maximum sections of the useful frequency range. In other words, velocity measurements might provide good sensitivity to change in condition of machinery over the broadest frequency range (68,96,97). However, other variables are also considered. Relative motion of the shaft can be measured in terms of displacement by a proximity (displacement) probe. The accelerometer seems to be the most popular contact transducer for measuring vibrations generated by components of gears and bearings because of its wide dynamic range. Given an acceleration or velocity signal, simple electrical integrating circuits are available to provide

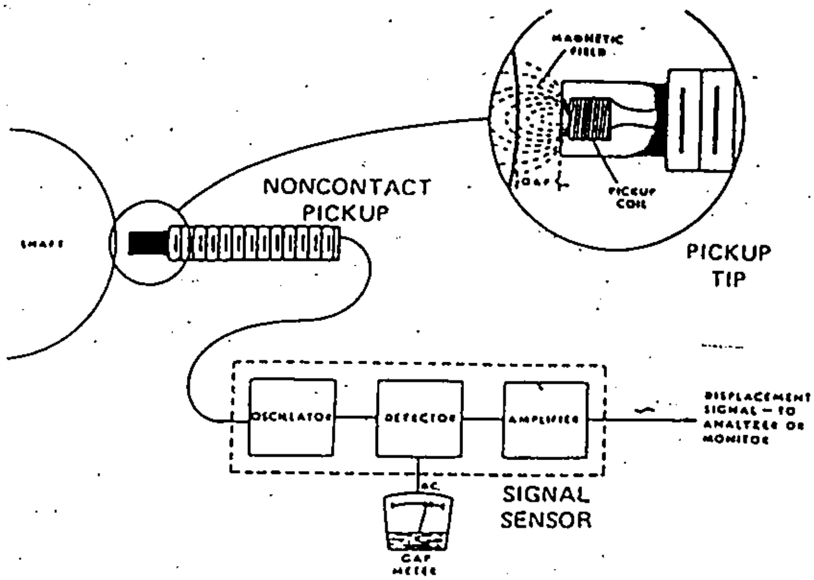


FIGURE 2.8: SCHEMATIC DIAGRAM OF NONCONTACT DISPLACEMENT PICKUP

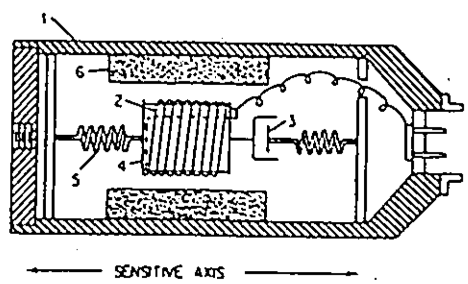


FIGURE 2.9: SCHEMATIC DIAGRAM OF VELOCITY PICKUP. (1) PICKUP CASE (2) WIRE COIL (3) DAMPLER (4) MASS (5) SPRING (6) MAGNET

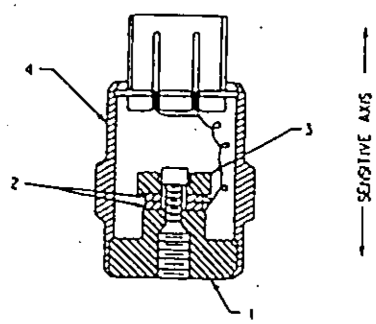


FIGURE 2.10: SCHEMATIC DIAGRAM OF ACCELEROMETER (1) BASE (2) PIEZOELECTRIC CRYSTALS (3) MASS (4) CASE

velocity and displacement. Jackson (84) fittingly stated that "velocity is the best, simple medium frequency parameter but the velocity pick-up is not necessarily the best type of transducer".

Perhaps the most important aspect to be considered is the frequency range over which a particular transducer is most useful (18,62). Measurements using non-contacting displacement probes are usually limited to an upper frequency limit of about 2000 Hz. This is due to the force limitations in the system being measured and the difficulty in eliminating surface imperfections which can be of the same magnitude as the values of displacement being measured at higher frequencies, rather than any inherent sensor limitation. Because velocity sensors are restricted by their construction to a useful frequency range of about 10 Hz to 1500 Hz, accelerometers have the broadest frequency response of all the commonly used vibration transducers. With the correct matching electronics, they are able to measure vibration at frequencies from below 1 Hz to as much as 50 kHz. Other factors of importance, such as dynamic range of measurement and transient response of the transducer, must also be taken into account. Perhaps the most important advantage of an accelerometer is its seismic arrangement, which provides an internal and effectively stationary reference point. This is a significant advantage if the transducer is mounted in a severe vibration environment.

One of the more popular methods of mounting contact transducer requires the use of magnets. It should, however, be noted that improper mounting leads to measurement errors that result in invalid data. The use of magnet or isolated mounting studs reduces the natural frequency of the transducers and consequently their useful frequency range. However, this does not mean that the mounting methods are improper but suggests that they must be used with discretion. Further discussion of characteristics of accelerometers and types of measurement error involved is available in the open literature (88,104,112).

2.4.2 Novel Sensors

Some of the other transducer types which are used in vibration monitoring will be mentioned briefly. A telemetry system (51) was developed to acquire vibration data from turbine blades and rotating shafts. Strain gages were used on blades to measure frequencies and amplitude of flow induced vibration.

Sattler (132) discusses a simple method for monitoring and measuring low level vibrations. It makes use of a magnetic phonograph cartridge to measure the vibration with minimal effects upon the motion

Philips (118-120) explains the use of an optical technique for detecting a faulty rolling element bearing. The deformation of the bearing races and the ball passage frequency are measured using a fibre optic probe. The rotational speed of the shaft is measured independently. A nominal design value of the ratio of the rotational speed to the ball passage frequency was established for a bearing in good condition. If the bearing is in good condition, the electrical output of the probe should result in a clean sine wave on an oscilloscope. A defective bearing will result either in a distorted sine wave or the shaft rotational speed to ball passage frequency ratio will deviate from the nominal design value.

An optical interferometric technique was developed to measure the amplitude and frequency of vibration in a gas turbine compressor blade. A laser is used as a coherent light source: the light illuminates two retroreflectors attached to the blades. The reflectors return the light to the source and mix it to form a pattern of fringes at the source. Relative motion between the retroreflectors produces a shift in the fringe pattern that is proportional to the deflection of the object carrying the retroreflectors (12).

A piezoelectric polymer sensor (58) was developed by the National Bureau of Standards from a polymer material that was made piezoelectrically active. The use of this sensor was also evaluated and compared to a piezoelectric accelerometer. Its advantages include high resonance frequency, the ability to be mounted to and conform to any shape and the ability to be mounted next to a vibration source.

2.4.3 Typical Equipment

Hand-held vibration meters, such as those manufactured by Bruel and Kjaer and IRD, have been used widely to provide overall vibration levels (as peak to peak or rms values). These are often sufficient to detect the presence of a machine fault. Earlier warning, however, can often be obtained by monitoring selected frequency bands using band pass filters. Such filters can be octave, one-third octave, 5% or 10% bandwidth based on the center frequency. Graphs of amplitude versus frequency can be manually drawn. Moreover, numerical data of this type can be stored in a computer for automated comparison with baseline data. Another logical approach is using a narrow band filter to gradually sweep through the frequency range of interest. The spectrum can be obtained by connecting the meter output to an X-Y plotter.

The Fast Fourier Transform (FFT) analyzer permits very fast and precise spectrum analysis. Typical analyzers provide 200,256,400, or 512 line frequency resolution to cover any one of its analysis ranges with what is equivalent to constant bandwidth filters. Thus, within a selected analysis range, frequency resolution is as good at high frequencies as at low frequencies. However, the upper frequency limit should be chosen only as high as required since the spectrum accuracy depends on the sampling time which should be longer for lower frequencies. The dominant frequencies will be enhanced by automatic spectra averaging. Several papers (61,69,70) discuss the concepts and applications of FFT analyzers.

Most new FFT analyzers have "zoom" features which provide a higher frequency resolution over any specified frequency range. This has significant value especially for identifying separate vibration sources even when they are close in frequency.

Narrow-band frequency analysis has been successfully used in diagnosing machinery faults in various industrial situations. Mitchell (106) compares spectra obtained with various filter bandwidth and also identifies the sum and difference frequencies due to meshing gears. The use of spectral analysis for detecting specific flaws in roller element bearings and in gears is discussed in detail by Taylor (140-144).

Time domain averaging, synchronized with the signal of interest, enhances it, while reducing all other signals. Hence, time averaging separates the signature of one mechanism masked by others. For example, if machine speed varies, analysis synchronized with shaft rotation produces the true spectrum of shaft vibration while other vibration data are reduced.

The FFT analyzer interfaced with a computer is capable of performing various statistical analyses such as "Kurtosis" which is the fourth moment of probability density function (56), crest factor which is the ratio of peak to rms values, and others (91,101).

The Shock Pulse Meter, which was developed in Sweden, is widely used to monitor the condition of roller element bearings. It is based on monitoring the mechanical impacts caused by bearing damage and operating problems. The differences between the base and peak levels are used to distinguish bearing faults from other sources of shock pulses (9).

The Vibration Spike Energy Meter developed by IRD, is also used to detect bearing failure (21,37). It measures the spike energy, which is produced by ultrasonic pulses in the microsecond range, caused by impacts between bearing elements which have microscopic flaws. Pulse amplitude, pulse rate and high frequency random vibration are electronically combined into a single quantity called g-SE (acceleration unit of spike energy) which is a measure of the bearing condition.

Another instrument is an envelope detector or a demodulator (developed by Shaker Research Corp.), which is basically an AM (Amplitude Modulation) detector. It produces a signal which follows the amplitude of the total signal but rejects the high frequency carrier. What results from envelope detection is a low frequency signal which is subsequently analyzed using a spectral analyzer.

Besides the above mentioned instruments, there are other useful instruments for both monitoring and diagnosis purposes:

1. A strobe light is used to determine the shaft rpm and the angular location of balance weights for a rotor in need of balancing;
2. A tracking filter is used to monitor the vibration component at running speed during start-up or shut-down of rotating machinery.
3. A spectrum tracking adapter is used to monitor vibration amplitude versus multiples of running speed. It is useful when the running speed varies slightly and automated comparisons are attempted.
4. A tape recorder is normally used to record the vibration signal for later analyses.
5. A digital computer is often used in the vibration monitoring system for the following reasons (33,92, 115,125):

a. automatic control of FFT analyzer functions;

b. storage of vibration spectra, process data, and machine descriptive information;

c. programmed instructions to operator with less chance for error;

d. flexible analysis presentation - CRT display, printout and X-Y plot;

e. automatic comparison with previous data which implies trend analysis;

f. capability for special studies to develop common signature patterns (includes baseline signatures), vibration limits, and diagnostics with automatic data processing;

g. improved problem solving via automatic data reduction, parametric analysis of vibration and process data and choice of graphic or digital presentation.

Furthermore, it provides flexibility for later incorporation of specialized input/output devices such as analog to digital converters for entering pressure and temperature data from the process and utility systems.

2.5 Data Analysis

A machine vibration signal measured in the time domain is generally indicated as an overall rms level. For many reasons, such a signal can be very complex and generally contains components from several sources: both wanted and unwanted. In addition, the measurement is often made at a point remote from the source of vibration, so the transmission path modifies the characteristics of the measurement. In some cases, such a measurement on a simple machine, which produces little or no harmonic content, a change in the overall rms level is sufficient to indicate a change in the condition of the machine. But this is not always true because changes in individual spectral content can indicate changes in the condition well before these changes produce a noticeable effect on the overall rms level. Hence, in most applications, there is a necessity for a more sensitive descriptor.

Figure 2.11 illustrates the result from a machine with a fault developing in one component. During five measurement periods, it is observed that the overall level remains essentially constant and then suddenly jumps up at the last measurement. Thus with the overall vibration level, the development of failure can not be predicted in most cases. However, following the continuing growth of the affected

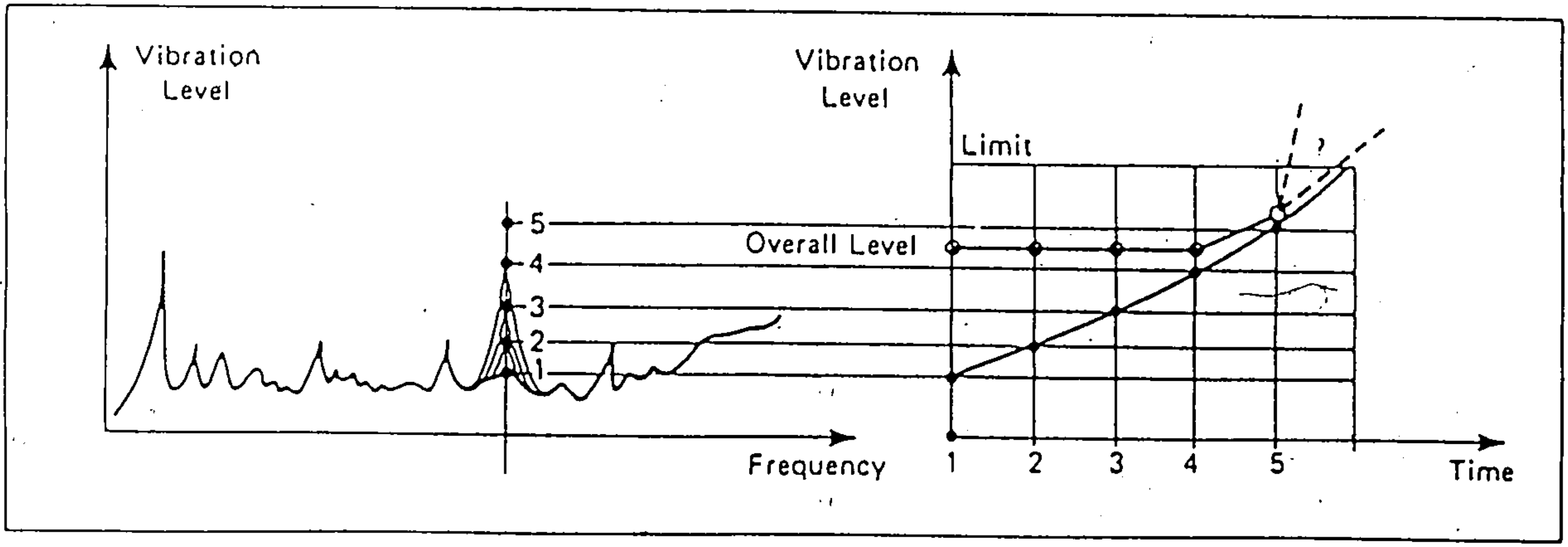


FIGURE 2.11: DEVELOPMENT OF A MACHINE FAULT AS SEEN BY OVERALL VIBRATION MEASUREMENTS AND BY SPECTRAL ANALYSIS (28)

frequency component over five measurement periods, the development of failure is clearly indicated, making possible the prediction of timing of required maintenance (28,108).

A frequency analysis is usually performed on vibration data. It is preferred because of the relative ease of correlating measurements and specific events in the operation of a machine (18,55,60,63,80,88,89,107,116,127,140-144). Questions often arise over the best format for data presentation and frequency resolution. The full octave band analyzer gives the lowest degree of frequency resolution. Glew (72) and Lundgaard (94) describe the use of an octave band analyzer to analyze and diagnose faults. Finer frequency resolution can be obtained with a smaller constant percentage bandwidth or a narrow band analysis. Most engineers prefer a discrete frequency band over a constant percentage bandwidth analysis; however, a conflict often arises between the need to cover a wide frequency band rapidly and fine frequency resolution.

Errors in computing frequency domain functions can lead to errors in data interpretation. Three useful short course notes (79,111,138) and several papers (11,61) on digital signal processing emphasize most of the sources of computational errors and ways to minimize them. For example, leakage occurs in any Finite Fourier Transform of a non-periodic signal resulting in distortion of the shape of the spectrum. The proper duration and window shape

(Hanning, Flat-Top, etc.) can reduce the unwanted effects.

As discussed earlier, vibration spectra of measurements from machines are complex because of the influence of transmission paths. Cepstrum analysis is a data processing technique that can be used to separate the periodic effects in a vibration spectrum. The mathematical definition of the Cepstrum is the power spectrum of the logarithm of the power spectrum. Note that the Cepstrum performs the same function as the autocorrelation function. However, effects multiplied in the power spectral analysis are additive in the Cepstrum analysis (20,126,127).

Considerable information regarding the condition of machinery is contained in the vibration signature up to 100 kHz. Many machine generate continuous high frequency vibration, modulated resonances, or periodic or non-periodic resonances. The modulation often indicates the presence of a fault. Through the use of filtering and envelope detection (demodulator), it is possible to extract diagnostic information. A powerful analysis technique, however, is the low frequency spectrum following the envelope detector. Burchill, Frarey, and Wilson (34,35) illustrate that the demodulated spectrum is fifty times as large as the low frequency spectrum. The main advantage of this technique is that the electronics are not driven to saturation by the high frequencies and the electrical noise found in many signals is not present. But most important of all, the equipment needed and the data processing are relatively simple and inexpensive. This technique has been used in monitoring the conditions of bearings and gears in many industries (29,59,66,81,151).

In reference (81), an evaluation of the effectiveness of various digital signal processing technique for determining the condition of mechanical components of helicopter power train is presented. Both, pattern and mechanically based diagnostic techniques are compared. Pattern recognition techniques are based on statistical treatment of data while mechanically based techniques rely on the dynamic characteristics of the system being tested. Various time domain and frequency domain procedures for indicating changes in the condition of bearings and gears are investigated. Mechanically based techniques show more promise as helicopter power train condition indicators because they are more flexible. The frequency domain is more popular for processing the data because specific events can be correlated with specific frequencies. Pattern recognition and trend analysis are implemented in a computerized monitoring system (137)

A digital computer-based technique was developed to program the variation of the intensity of a printout on a line printer as a function of the power spectral density at each frequency. The technique was used to investigate the effects of operating conditions and failures on the vibrations of a cam and follower mechanism. The technique can also be applied in determining the condition of roller element bearings (153).

Until recently FFT analyzers were in such widespread use that engineers tended to rely solely on the frequency domain for vibration analysis. However, valuable information can be obtained from the time domain. Fortunately, its advantages are becoming more widely recognized. Direct differential time measurement is given as a digital readout on some of the new analyzers and oscilloscopes. Such readouts allow accurate identification of the period of vibration events.

The use of time domain averaging for detecting periodic signals is quite extensively employed. Data processing in the time domain can be used to identify the source of vibration and the condition of certain types of rotating machinery (78,140,144). However, vibration signals from most rotating machines are not completely periodic. Braun and Seth (23) introduce the concept of a combination of time domain averaging and variance analysis. It enables the extraction not only of periodic but also of random transients, time locked to the rotation. Thus a unique detailed description of the signal which is not available from regular spectral analysis can be obtained. Digital filtering in the frequency domain was also proposed for extracting periodic signals from rotating machinery. However, it applies only to leakage free signals. The use of appropriate windows is suggested for signals that are not coherent with the rotational frequency of the machine, in order to minimize errors.

Braun (24) also proposed another approach for extracting weak periodic signals from periodic signals, generated by rotating machinery, by computing changing variances. It is based on a combined computation of averages and variances. It is done in real time; hence, the running variance is computed instead of the time variance. If sufficient samples are taken, the running variance approaches the true variance. This technique has been extended to non-stationary vibration signatures.

Dyer and Steward (56) developed a statistical vibration analysis technique for detection of roller element bearing faults. This method is based on the fourth statistical moment, or Kurtosis, that remains constant for an undamaged bearing irrespective of load and speed, yet changes with damage. The extent of damage can be assessed from the distribution of this statistical parameter in selected frequency ranges.

Another technique, for detecting faults in rotating machinery is based on the cumulative distribution function of maximal and minimal magnitudes (peaks) in an acceleration-time history in specific amplitude bands(41).

Piety and Magette (121-123) formulated and later implemented statistical techniques for automating the detection of anomalous performance of rotating machinery. The effectiveness of the computer-based vibration monitoring system was evaluated on a small rotor assembly in laboratory tests. Vibration signals from both displacement probes and accelerometers were obtained. Time and frequency domain descriptors were selected to compose an overall signature that characterized the monitoring equipment. Limits for normal operation of the rotor assembly were established automatically during an initial learning period. Then anomalous detection was accomplished by applying an approximate statistical test to each signature descriptor.

Digital computers are widely used in various industries for performing not only the data processing function but also the command and control functions for monitoring the condition of rotating machinery. In most cases, mini/microcomputers are used. The primary reason for the recent effort to apply mini/microcomputers to machinery condition monitoring has been the substantial reduction in their cost. Other reasons include high flexibility and varied capabilities and also a lower cost than the hardwired individual vibration monitors and alarms of many systems. A major limitation of the mini/microcomputer based vibration monitoring system is reliability, hence, backup monitoring and alarm systems have to be provided for very critical machinery (75,76).

2.5.1 Bearings

Ball and roller bearings generally follow a characteristic wear-failure pattern. Owing to the low wear rate and high roller rate contact load, they often tend to fail by fatigue rather than by wearing out (22). Figure 2.7 shows a simplified but typical wear rate curve for a rolling element bearing. Here the wear rate is proportional to the vibration level. High wear rate is a characteristic of the run-in period. Then there is a long period with low wear-rate after which surface defects start to appear. The main causes of defects are lubrication failure, fatigue, dirt, corrosion, electric discharges between ball and races, brinnelling, excessive preloading, vibrations, incorrect assembly/mounting and high temperature (22,95). Monitoring the condition of ball bearings requires the determination of the point at which the rate of change in vibration level begins to increase.

The characteristic frequencies of rolling element bearings are a function of geometry and rotational speed. Equations for predicting these frequencies are available (140-142,144). Each bearing component has a resonance frequency dependent on geometry, loading, and bearing material. These frequencies which are higher than rotational frequencies are excited when a component encounters a flaw in another component, e.g., a ball passing a flaw in the inner race.

There are numerous techniques which have been developed to determine the condition of rolling element bearing. One of the more popular techniques for detecting bearing failures is spectral analysis. This technique is capable of predicting failure in bearings by using vibration signature data and performance history. The success of this technique in the detection of ball bearing failures is well-documented in many technical papers (1,90,100,116,140,144).

The sensitivity of various parameters, such as overall vibration level, spectral analysis, detected carrier spectrum analysis, carrier rms level and amplitude distribution of the carrier signal for determining the condition of rolling element bearings, was investigated. Data were obtained on good and defective (implanted defects) rolling element bearings. The high frequency techniques, particularly the detected carrier spectrum analysis (envelope detection) were the most sensitive to deterioration in the condition of the rolling element bearings (4,5). Other evaluations of the sensitivity of this technique to rolling element bearing damage are also available (20,34,35,48,58,66,81,114,149).

Dyer and Steward (56) employed a statistical vibration analysis for detection of rolling element bearing damage. The technique is based on a statistical parameter, Kurtosis. It is capable of detecting the earliest onset of failure i.e. point B in Figure 2.7, thus detecting bearing deterioration earlier than the measured peak or rms acceleration techniques. The feasibility of using signal decomposition techniques for determination of bearing condition was evaluated (22-26). This technique claims to identify defects in the outer race of a rolling element bearing that could not be detected by spectral analysis (22).

Another technique, used for detecting rolling element bearing defects, involves ultrasonic diagnostic procedures. Catlin (37) and Braithwaite (21) discuss the mechanisms which cause ultrasonic signals (spike energy) in defective bearings and the parameters which affect the characteristics of these signals. The signal amplitude, signal rate and high frequency random vibratory energy are electronically combined into a single quantity, g-SE, which is acceleration units of spike energy. IRD developed the spike energy meter which is also commonly used in many industries (8,10,21,37,68).

The Shock Pulse Method is a relatively new field of monitoring the condition of rolling element bearing. It is based on high frequency acceleration signals, referred to as shock pulses (30-50 kHz), as a measure of condition of bearings. This technique has shown some success in detection of bearing damage (9,67,91).

Crest factor analysis is yet another method of determining the condition of rolling element bearing by indicating the size of a fatique spall. It is based on the peak to rms ratio of the processed signal; the signal increasing with spall size (91,101).

Philips (118-120) evaluated an optical technique for detecting damaged bearings by monitoring an electric motor bearing. He claimed that this technique gives a better and earlier indication of bearing damage than the use of an accelerometer. The main advantage of this method is that the data obtained are not confounded by the resonances and vibrations of the surrounding structures.

Rogers (130) describes the detection of incipient failures of rolling element bearings by Kurtosis and the location of fatigue cracks in slowly rotating bearings by acoustic emission. Acoustic emission is the term applied to the spontaneously generated elastic waves produced within a material under stress. Plastic deformation and the nucleation and growth of cracks are the primary sources of acoustic emission in metals. These acoustical signals can be detected by remote sensors and their source can be located by comparing the arrival times at several sensors. Thus, by "listening" to the structure crack growth can be detected, located and monitored, giving thus a warning of incipient failure.

2.5.2 Gears

Vibration signatures generated by gears are generally periodic with the meshing frequency: the product of the number of teeth and shaft speed. The meshing of two teeth results in impacts even if the gears are operating normally. Gear defects such as worn and chipped teeth can be distributed due to pitch diameter errors and shaft eccentricity. Such faults cause the development of sidebands at several orders of rotational speed of the shaft.

Interpretation of vibration signatures in terms of gear condition is often more difficult than the detection of gear faults because many faults appear at the gear mesh frequency. It was observed that one type of gear defect might excite certain types of resonances in the gear housing or machine structure, but other defects might excite entirely different resonances.

Several techniques, including observation of time history signal, spectral analysis, and sum and difference frequencies, have been used to identify gear defects. Proper location of vibration transducer is essential in order to identify the source of a gear defect. Taylor (142,143) performed careful analysis of the time signal and spectrum frequencies, shape, amplitude, and sum and difference frequencies. Consequently, he determined the number of defective teeth on a gear, the number of gears with defective teeth, and the location of the defective teeth relative to each other and to some datum point.

2.6 Users' Experiences

Vibration monitoring is presently practised on many different levels of sophistication in a wide range of industries. Recent surveys (147) of machinery vibration monitoring in various industries indicate that the future trend is towards automated (computer-based) vibration monitoring systems with fault diagnosis capabilities.

By far the simplest type of vibration monitoring system is the measurement at regular intervals of the overall root mean square levels of velocity or acceleration at specific point on the casing of selected machinery. The vibration level is used as an indicator of the condition of the machine. Comparison of the measured level with values on the standard vibration severity chart allows the assessment of the machine (4,40,145). This system is useful but it provides only a limited amount of diagnostic information.

Another approach involves measurement in octave bands. It is used by the Canadian Forces (72,74) to check the quality of repairs and overhauls of machinery as well as to perform acceptance tests on new machinery in ships. On aircrafts, it is used for special task such as checks on the alignment of engine drive trains and gearbox vibration levels. It was found that octave band analysis is sufficient to define the nature of many problems but that it is not always sensitive enough to identify the defective element.

A similar approach is also used on selected U.S. Navy surface ships (148).

Most organizations undertake vibration signature analysis as a logical extension or refinement of an ongoing program such as the two mentioned above. Other organizations evolve into vibration signature analysis from a continuous monitoring in an effort to gain more insight into the machine condition so they can anticipate rather than react to problems. Still others find periodic vibration signature analysis the most cost effective means to monitor the condition of critical smaller equipment, where installed sensors, hardwired into a central monitoring system, may be prohibitively expensive.

There are two basic methods available for accomplishing periodic vibration signature analysis on operating machinery. Some companies favor recording vibration data on magnetic tape for later analysis in a laboratory (20,53). Other companies, on the other hand, house the monitoring equipment in a trailer or van which can be brought on site for direct data acquisition, reduction, and analysis (36,93,135). Modern versions of both methods are based on an FFT analyzer interfaced to a desktop computer.

The petroleum industry has had extensive experience with vibration signature analysis for monitoring the condition of rotating machinery. Results reported from a chemical plant show that it is possible to use on stream vibration measurements to detect small defects in bearings and gears, loose parts or faulty operation (15,16,85). Significant cost savings were achieved at a refinery where a vibration monitoring system was employed (82). Unscheduled shutdowns due to catastrophic failures were reduced, running time of machinery was extended, and the need to tear down and inspect machinery was reduced. The scope of the program expanded from measuring unfiltered bearing cap vibration levels on a few components of minor machinery to measuring vibration on all machines and the use of sophisticated equipment to diagnose faults. An updated report (49) on this vibration monitoring system describes a total approach applied to all machinery measurements at this refinery. It is claimed that a thirty percent cost saving has been achieved since the start of the vibration monitoring program.

There is a great deal of published literature (3,19,92, 103,104,109,110,129) on vibration monitoring systems incorporating a computer as a method of saving manpower. In this scheme, the computer accomplishes the comparison with the baseline data and prints out any deviation outside a predetermined normal envelope. Using the computer, data can also be modified, processed, saved, and recalled allowing a very high degree of flexibility. This approach frees employees from the tedious and not particularly interesting task of comparing data so they will have more time to concentrate on analyzing problems.

Bradshaw and Randall (20) describe the practical experience gained with a vibration monitoring system as used on gas turbines, pumps, and power generation units on the Trans Alaska Pipeline. The system consists of an FFT analyzer interfaced to a desktop computer and analyses are made from tape recordings of vibration signals obtained using accelerometers and taken at intervals of approximately three months. The data analysis techniques involve detailed frequency analysis because of the complexity of the machines and the limited number of measurement points. The system claims to predict and diagnose a wide range of incipient faults on high speed rotating machines. In one case, the system detected and diagnosed a developing outer race fault in a ball bearing about eight months prior to repair.

The most complex method for accomplishing vibration signature analysis is to have the vibration transducers permanently installed and hardwired to a central location where the analyzing equipment is situated. This is the most expensive approach, however, it has been found to be cost effective for monitoring a large number of rotating machines in various industries. Almost all of the automated (computerized) vibration monitoring systems are specially designed to suit each particular industry. The data acquisition hardware, minicomputer and signal conditioning hardware are all chosen to meet the specific needs of various industries. Even the software has to be written specifically to satisfy each requirements. Hence, there is no one particular computerized vibration monitoring system capable of satisfying all industrial applications.

Bultzo (33) describes the installation of a minicomputer based vibration monitoring system in a chemical plant that achieved a 100 percent return on the investment in the first year of operation. The return is based on process credits resulting from downtimes that are either shortened or postponed and parts and manpower savings. The system measures trend and stores measurements of many parameters, thus, providing insight into the vibration of machinery. The system also uses off-the-shelf hardware and has been highly reliable. Software is specially written to satisfy each specific need. Furthermore, because of the successful operation of the vibration monitoring system, two additional systems were justified and consequently purchased for the plant. Other chemical plants have also implemented computerized vibration monitoring systems for on-stream condition monitoring of critical machinery (15,16,42,125). Since these systems have been in operation, only a few minor repairs have had to be made. A computer coupled with an FFT analyzer for automated analysis of machinery vibration data was found to be flexible both as a diagnostic tool and for condition monitoring (32,82,115).

One nuclear power plant (77) implemented the latest approach to vibration monitoring utilizing microcomputer based data acquisition stations and the computer network approach to communications between the control computer and the stations. The data acquisition stations replace the vibration meters and allow greater flexibility in setting alert and alarm levels and also eliminate the need for manual data collection. The control computer permits monitoring of machinery from a central location and provides computational speed to perform sophisticated analysis on vibration data and the storage space for long term trend and signature analysis. The network linking the data acquisition stations to the control computer permits the vibration information from all machinery in the plant to be analyzed, displayed, and stored at a central location.

The pulp and paper industries are also involved in the use of automated vibration monitoring systems. One system is used solely to monitor the condition of bearings in large paper machines (108). Patenaude and Axelsson (117) describe ASEA Ltd.'s experience with the computerized monitoring system for paper mills. The system monitors all digital and analog signals (including vibration) and registers alarm signals from the process. To facilitate fault tracing, the signals can be processed to generate indications of events at suitably located display units.

The final issue is the cost of the vibration monitoring system for monitoring the condition of machinery as well as diagnosing machinery faults. The potential payoff of monitoring machinery condition has been demonstrated by users who have found it cost effective: vibration monitoring has also prevented costly unscheduled outages. The costs of transducers, signal conditioning equipment, data processing equipment, readout equpment, and other specialized measuring equipment must all be considered for the most effective system. Many vibration monitoring systems are exceedingly expensive because of duplication of instrumention, minicomputers and relatively expensive sensors. Thus, an extensive knowledge of the machinery to be monitored coupled with the limitation of the instrumentation and its price, aid in the design of an optimum vibration monitoring system.

III. THEORY

One of the most common signal processing techniques used for machine health monitoring is spectral analysis which utilizes Fourier Analysis.

The discussion on Fourier Series and Discrete Fourier Transform are omitted because they are generally standard theories which can be found in textbooks.

But however, there are two related topics that have to be discussed: the Fast Fourier Transform Analysis and the Bearing Frequency Analysis.

3.1 <u>Fast Fourier Transform Analysis</u>

The structural dynamic analyzer is one of the main pieces of equipment used in analyzing the vibration data. It employs the Fast Fourier Transformation which is an algorithm for efficiently transforming data from the time domain to the frequency domain based on the Discrete Fourier Transform.

There are many factors which complicate the use of the Fast Fourier Transform. The Transform must be implemented on a digital computer if the results are to be obtained quickly and accurately because of the many calculations involved in transforming domains. Additionally, we cannot transform from time to frequency domain in a continous manner but instead must sample and digitize the time domain input. This implies that the FFT algorithm transforms digitized samples from the time domain to samples in the frequency domain. Furthermore, because we have sampled, we no longer have an exact representation in either domain. However, a sampled representation can be as close to ideal as we desire by placing the samples closer together.

The operational schematic of a Fast Fourier Transform (FFT) Analyzer is illustrated in the block diagram of Figure 3.1. Analog inputs are first passed through an anti-aliasing low pass filter prior to Analog to Digital (A/D) Conversion. The sampled data are digitally filtered and then placed in an input digital memory and prepared for processing. Before being operated on by a Fast Fourier Transform algorithm, data samples in input memory are typically time-windowed and any other necessary bookkeeping functions are performed.

The FFT analyzer allows rapid computation and display of the frequency spectrum and power spectrum as a function of frequency. However, certain precautions must be taken and assumptions used in the procedures must be understood in order to achieve correct data interpretation. This is particularly true whenever data has transients or varies in its composition with time.

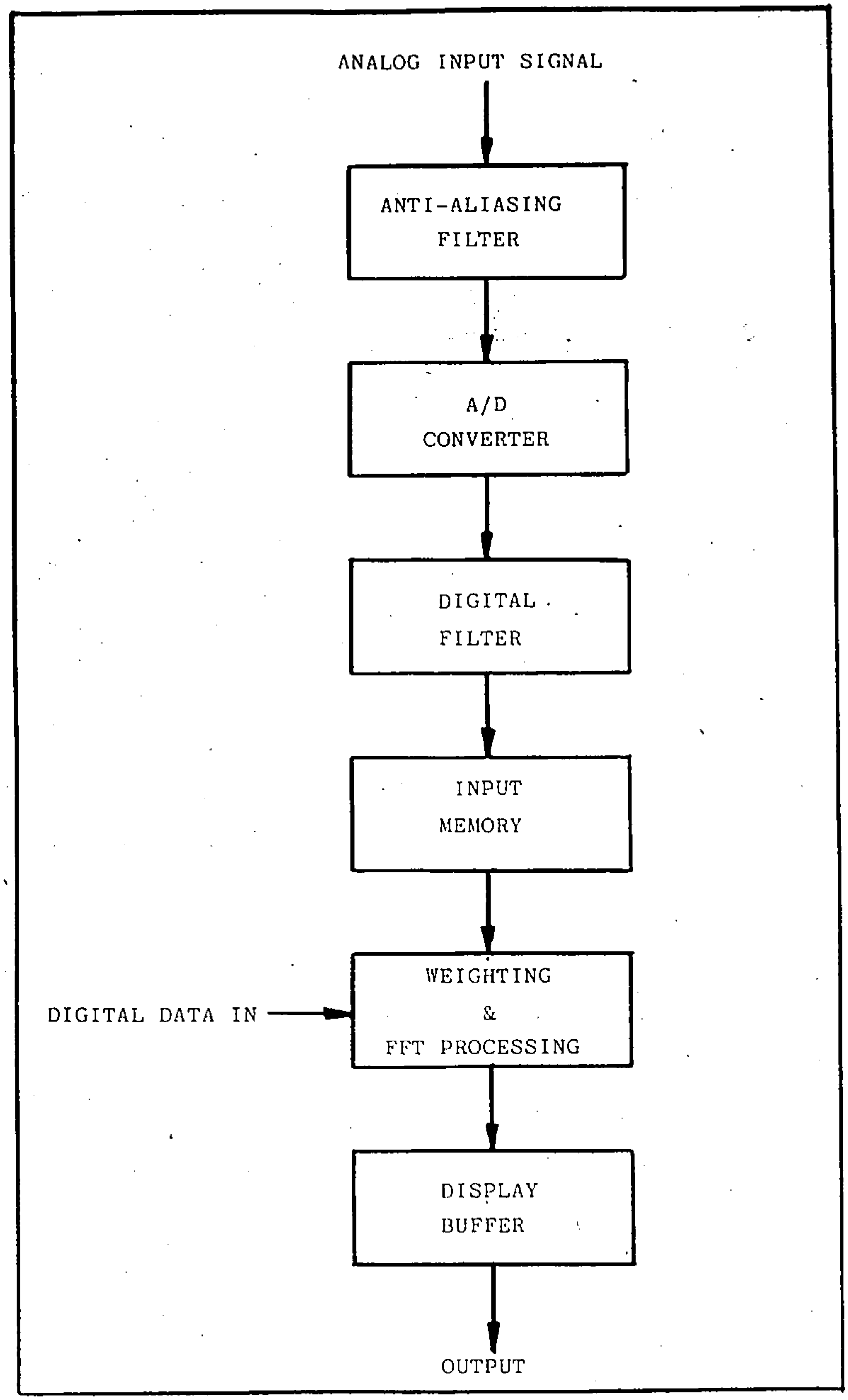


FIGURE 3.1: FAST FOURIER TRANSFORM ANALYSIS CONCEPT

3.1.1 <u>Sampling</u>

There are three main areas that have to be discussed in sampling; the time record length, the sampling frequency and the frequency resolution.

A time record, as depicted in Figure 3.2, is defined as K consecutive, equally spaced samples of the input. Because it makes the FFT algorithm simpler and much faster, K is restricted to a multiple of 2, for instance K = 1024 for the Hewlett Packard 5423A, Structural Dynamics Analyzer. This time record is transformed as a complete block, into a complete block of frequency lines. All the samples of the time record are needed to calculate each and every line in the frequency domain. This is the block processing property of the analyzer.

Thus, there cannot be valid frequency domain results until a complete time record has been gathered, because the FFT transforms the entire time record block as a total. However once completed, the oldest sample could be discarded, all the samples shifted in the time record, and a new sample added to the end of the old time record as in Figure 3.3.

The longer the total time record length T, the more accurate is the frequency resolution capability. In fact,

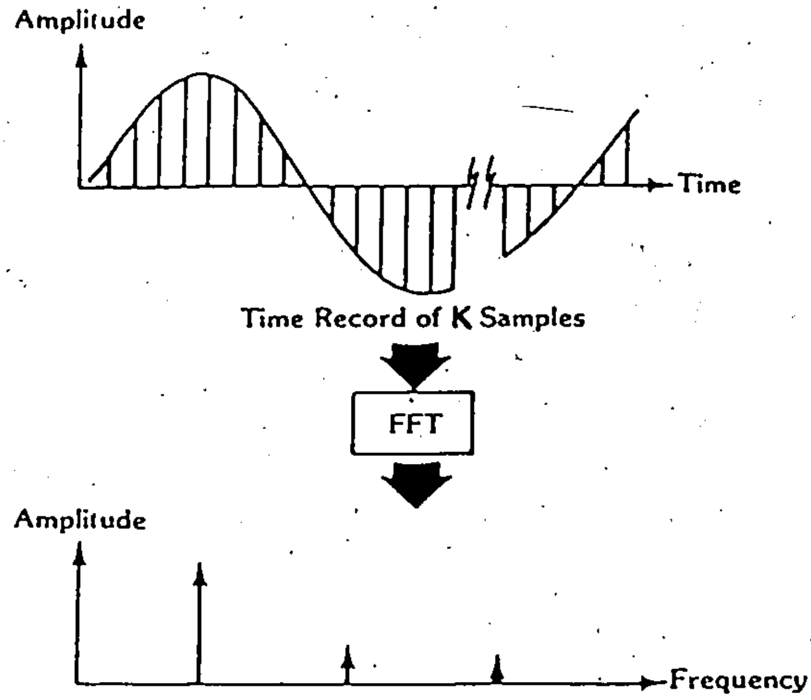


FIGURE 3.2: FFT WORKS ON BLOCKS OF DATA

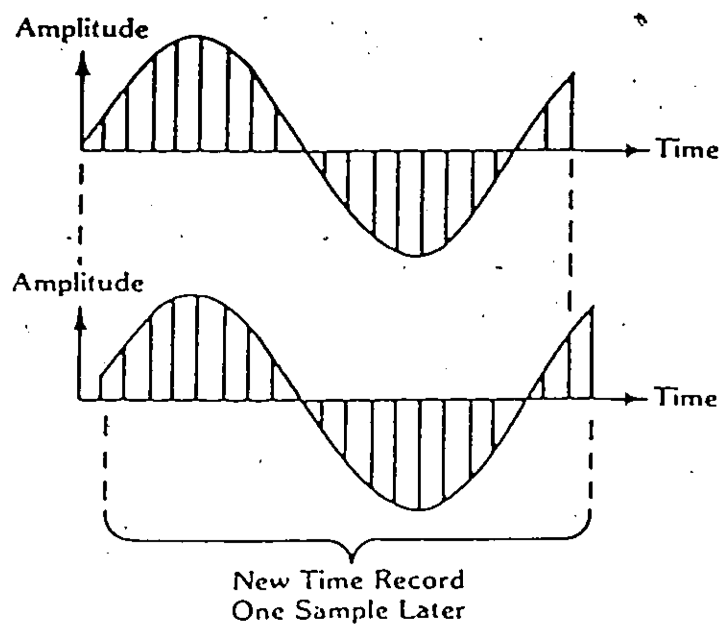


FIGURE 3.3: A NEW TIME RECORD EVERY SAMPLE AFTER THE TIME RECORD IS FILLED

according to Rayleigh's Criterion, the frequency resolution is inversely proportional to the total signal length, i.e.,

$$\Delta f = \frac{1}{T} \tag{3.1.1}$$

FFT analyzer requires digitized samples of the analog time signal for its digital calculations. Thus the input signal must be sampled often enough to adequately characterize the signal in the frequency domain. Nyquist Sampling Theory states that the sampling frequency must be at least two times the highest frequency of interest, i.e.,

$$f_S \geqslant 2\, f_{max} \tag{3.1.2}$$

The interrelationship of sampling parameters is given by the following equations:

$$\Delta t = \frac{1}{f_S} \tag{3.1.3}$$

Assuming, $f_S = 2\, f_{max}$, then

$$f_{max} = \frac{1}{2\,\Delta t} \tag{3.1.4}$$

$$T = K\,\Delta t \tag{3.1.5}$$

$$\Delta f = \frac{2 f_{max}}{K} \tag{3.1.6}$$

$$BW = f_{max} - f_{min} = \frac{K\,\Delta f}{2} \tag{3.1.7}$$

where:

Δt is time interval between digital history values,

T is total time record length

f_S is frequency of ADC sampling operation,

f_{max} is maximum frequency of interest,

f_{min} is minimum freqeuncy of interest,

BW is bandwidth (frequency range of interest),

Δf is frequency resolution,

K is number of samples in a time record

3.1.2 Aliasing

The reason an FFT analyzer requires so many samples is to avoid a problem called aliasing. Aliasing is a sampling error caused by processing data with a frequency content that exceed the highest frequency allowed by the sampling rate. Generally aliasing results from sampling too slowly or having signal energy outside the frequency range of interest (see Figure 3.4). Two signals are said to be alias if the difference of their frequencies falls in the frequency range of interest. This difference in frequency is always generated in the process of sampling. In Figure 3.5, the input frequency is slightly higher than the sampling frequency thus a low frequency alias term is generated.

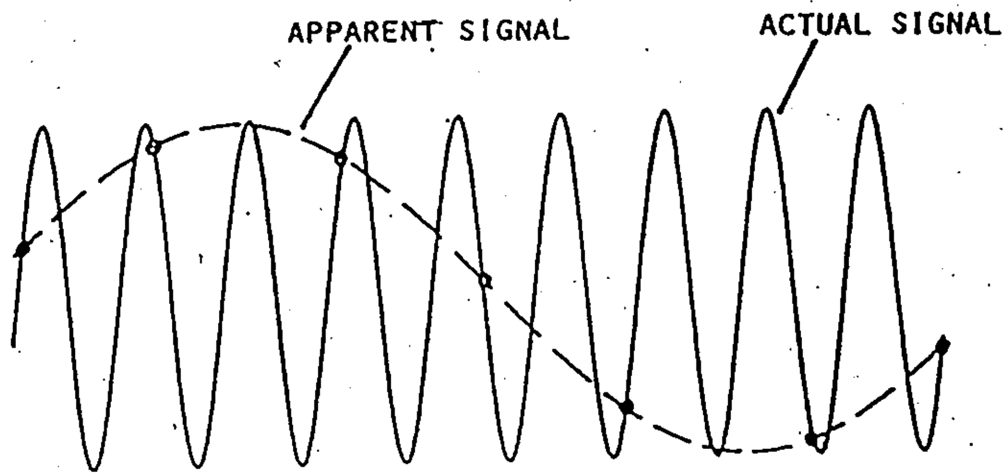


FIGURE 3.4: ALIASING RESULTING FROM SAMPLING TOO SLOWLY OR FROM HAVING SIGNAL ENERGY OUTSIDE THE BAND

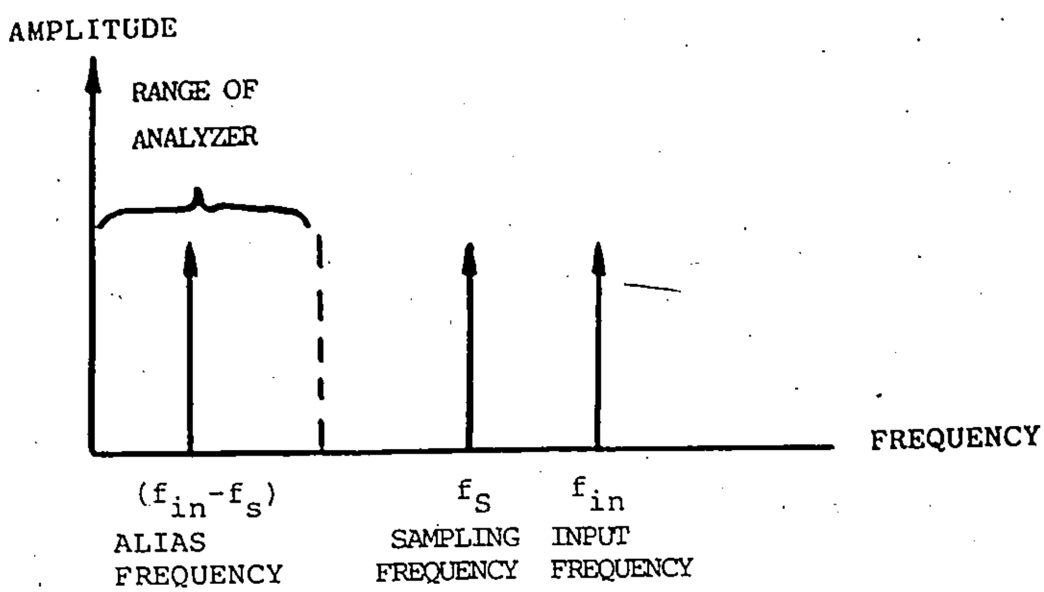


FIGURE 3.5 : THE PROBLEM OF ALIASING VIEWED IN THE FREQUENCY DOMAIN

As long as we sample at greater than twice the maximum frequency of input, the alias products will not fall within the frequency range of input, as illustrated in Figure 3.6.

Another method to reduce aliasing error is the use of an anti-aliasing filter. An ideal anti-alias filter is illustrated in Figure 3.7a. It would pass all the desired input frequency with no loss and completely reject any higher frequencies which otherwise could alias into the input frequency range. Theoretically, it is impossible to build such a filter. Instead most filter characteristics look like in Figure 3.7b with gradual roll off and finite rejection of undesired signal. Large input signals which are not well attenuated in the transition band could still alias into the desired frequency band. Thus sampling frequency must be greater than twice the highest frequency of the transition band, as illustrated in Figure 3.8.

In the analog case, a new filter has to be used when sample rate of the ADC changes. When using digital filtering, the ADC sample rate remains constant at the rate needed for the highest frequency range of the analyzer. This implies that we do not have to change our anti-aliasing filter. Following the ADC is a digital filter to obtain the reduced sample rate and filter we need for the narrower frequency span. This implies that only one anti-aliasing filter is required - a significant savings in equipment (see Figure 3.9).

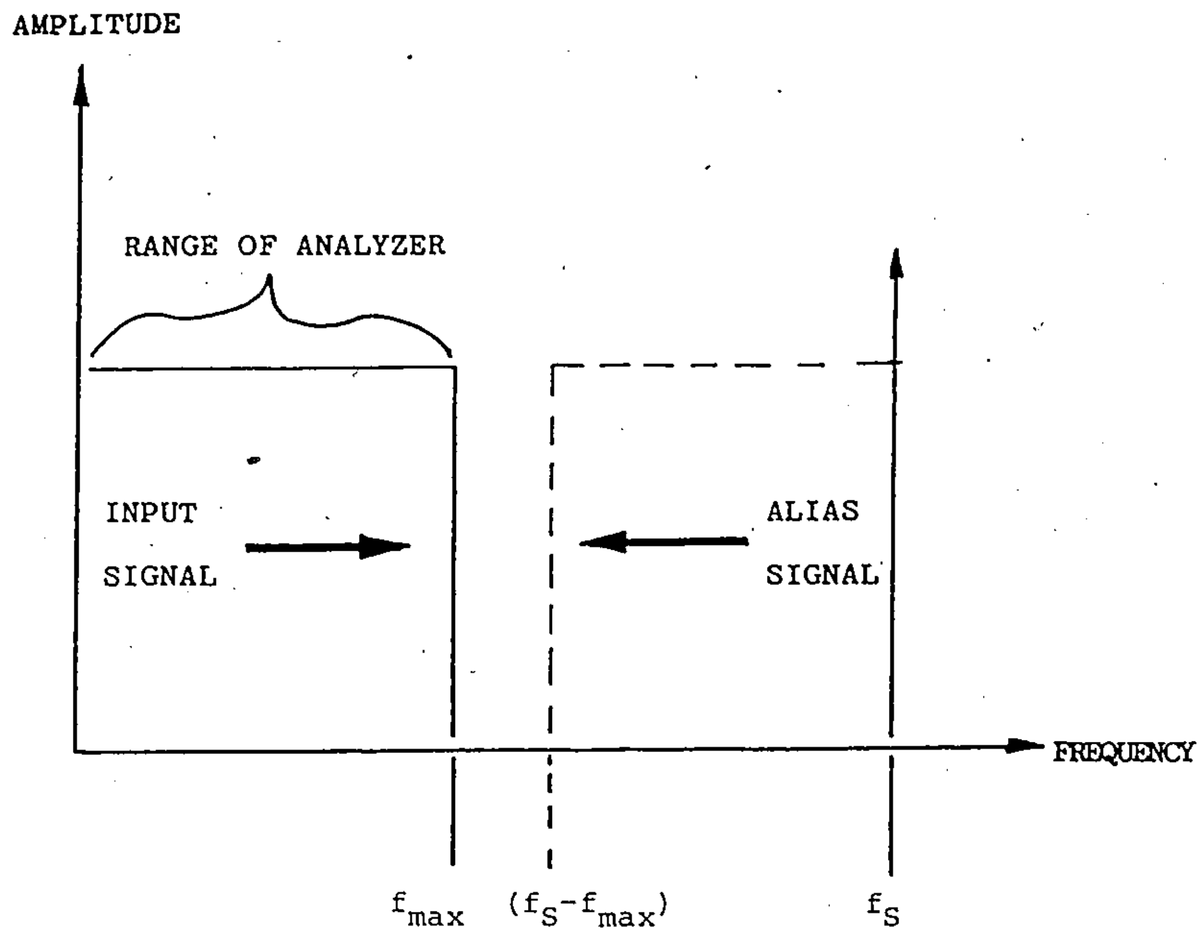


FIGURE 3.6: A FREQUENCY DOMAIN VIEW OF HOW TO AVOID ALIASING — SAMPLE AT GREATER THAN THE HIGHEST INPUT FREQUENCY

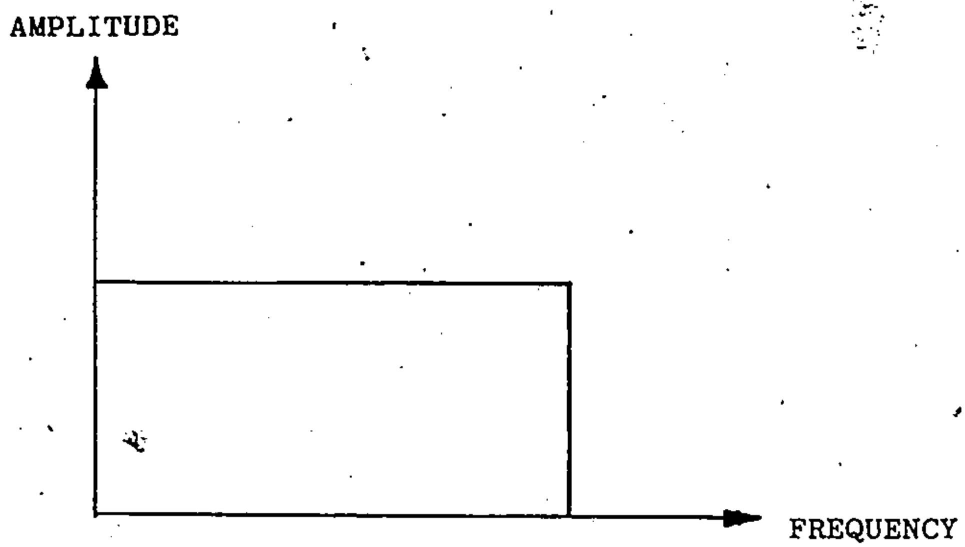


FIGURE 3.7a: "IDEAL" ANTI-ALIASING FILTER

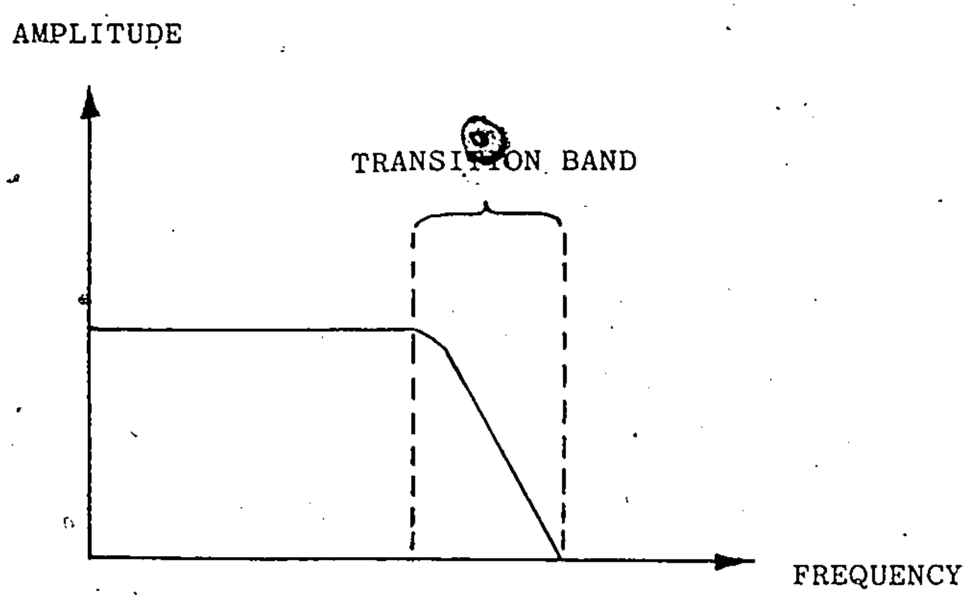


FIGURE 3.7b: "REAL" ANTI-ALIASING FILTER

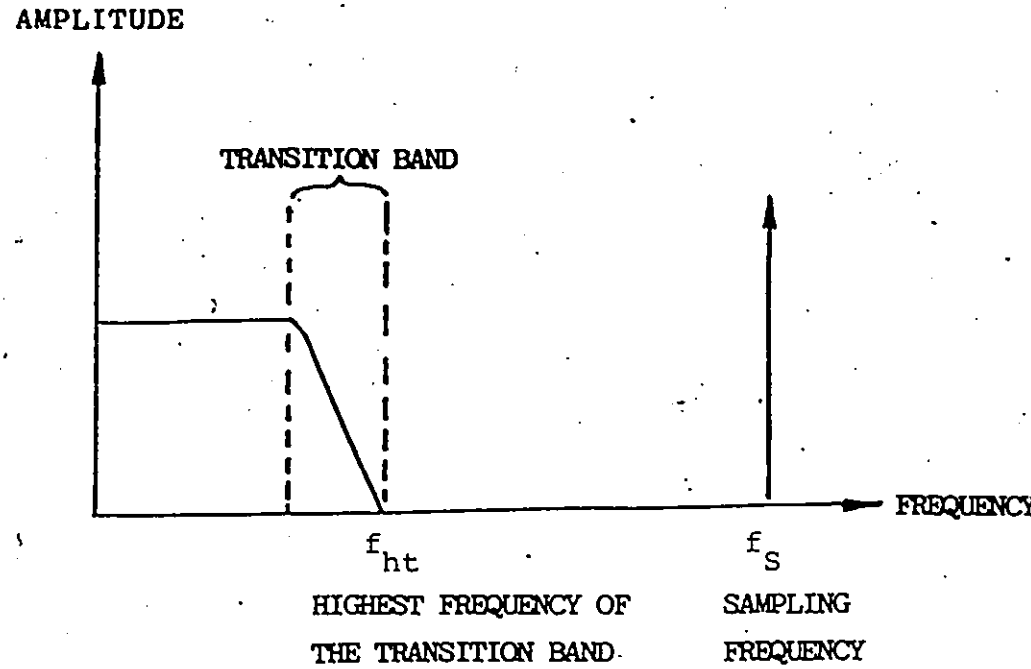


FIGURE 3.8: ELIMINATION OF ALIASING ERROR BY USING ANTI-ALIASING FILTER AND SELECTING SAMPLING FREQUENCY

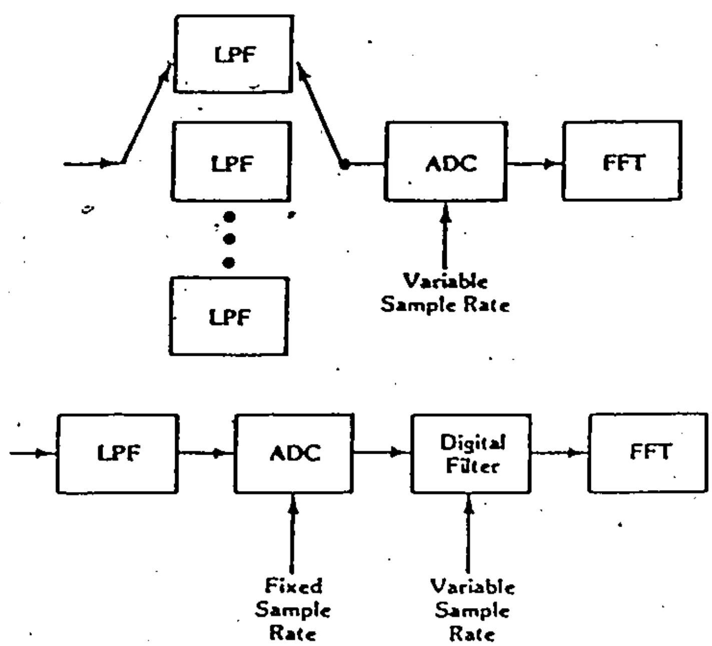


FIGURE 3.9: BLOCK DIAGRAMS OF ANALOG AND DIGITAL FILTERING

3.1.3 Quantization

Quantization is a process whereby the amplitude of the analog signal is converted into discrete amplitudes. Typically, the amplitude range is divided into many intervals as shown in Figure 3.10. The number of intervals is determined by an Analog to Digital Converter. Most often, the amplitude is represented by a binary number. A 12 bit A/D converter (as in the HP 5423A) divides the amplitude range into 4095 intervals resulting in a resolution of approximately 0.025 percent of full scale.

Emerging from the A/D Converter is a string of numbers representing the amplitude of the input signal at sampling instants. Although the Nyquist Theory states that there must be at least two samples per cycle of the highest frequency component to be analyzed, most spectrum analyzers use a sampling rate of 2, 2.56 or 4 times the highest frequency component.

If the sampling frequency is 4 times the highest frequency component, a 25 kHz spectrum generates 100,000 words per second. At this rate, a 64 k word computer is completely filled in approximately 0.5 second.

Thus it is unrealistic to save all the samples coming from the digital filter. Instead, an input memory stores a smaller number -- typically, 1024 samples. At any time,

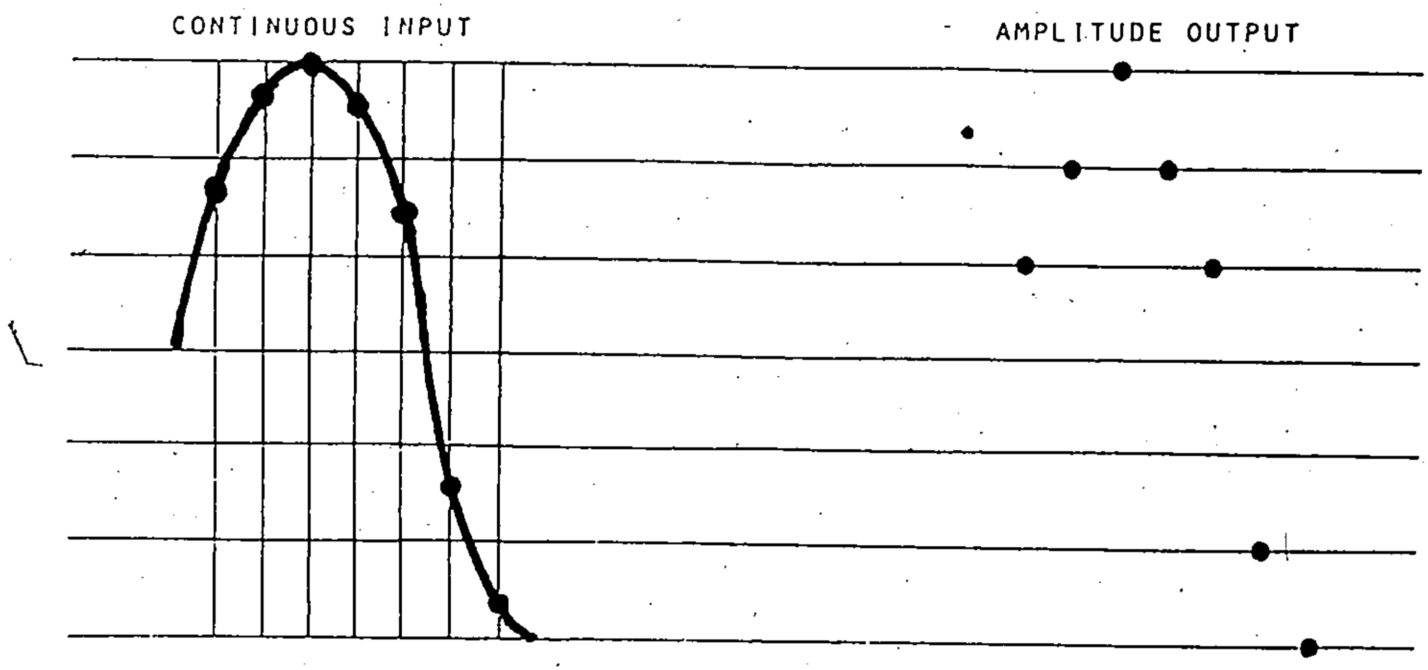


FIGURE 3.10: ANALOG INPUT CONVERTED TO DISCRETE AMPLITUDE

the contents of this memory can be read and the latest data displayed. The size of this memory determines the time record length. Because the buffer has a fixed capacity, K, the relationship of the frequency resolution to the maximum frequency is fixed.

If the sampling rate is 2 times f_{max} then from Equations (3.1.3) to (3.1.6), we have

$$T = \frac{K}{2\ f_{max}} \tag{3.1.8}$$

For a buffer size of 1024,

$$T = \frac{1024}{2\ f_{max}} = \frac{512}{f_{max}}$$

From Equation (3.1.1),

$$\Delta f = \frac{f_{max}}{512}$$

Since sampling is done at a higher rate for higher analysis ranges, the time record length T, decreases. At lower analysis ranges the time record length increases.

The FFT Processor accepts the K word block from the input buffer and performs Fourier Analysis on the data to determine the frequency and amplitude of the spectral components.

Just as the input signal to the processor was discrete (time samples and amplitude quantization) so too is the spectrum discrete (sampled frequency intervals and quantized amplitude). This discrete property is the result of performing a Discrete Fourier Transform on the input data block.

The processor cannot determine frequency components at every conceivable frequency in the band because to do so would require infinite memory. Instead the analysis band is described at 512 locations (HP5423A using Hi-Res Auto Spectrum). Other modern analyzers may have different numbers of discrete locations.

3.1.4 Leakage

The FFT algorithm assumes the input signal to be periodic with respect to the time record length T, as illustrated in Figure 3.11. While Figure 3.12 shows the effect of input signal not periodic with respect to the time record length T. This phenomenon which causes the smearing of energy throughout the frequency domain, is known as leakage. The causes of leakages are finite time record length and Discrete Fourier Transformation.

If the input frequency was exactly at one of the analyzer spectral lines, the spectrum of this signal when viewed on the analyzer would be a single point. This single point output would occur for any input frequency

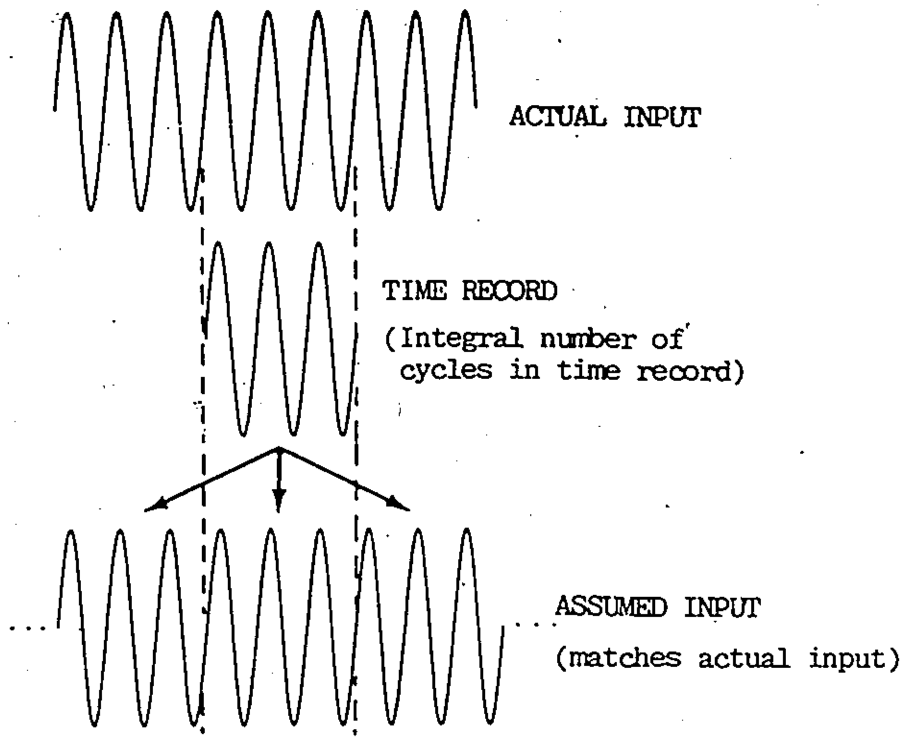


FIGURE 3.11: INPUT SIGNAL PERIODIC IN TIME RECORD

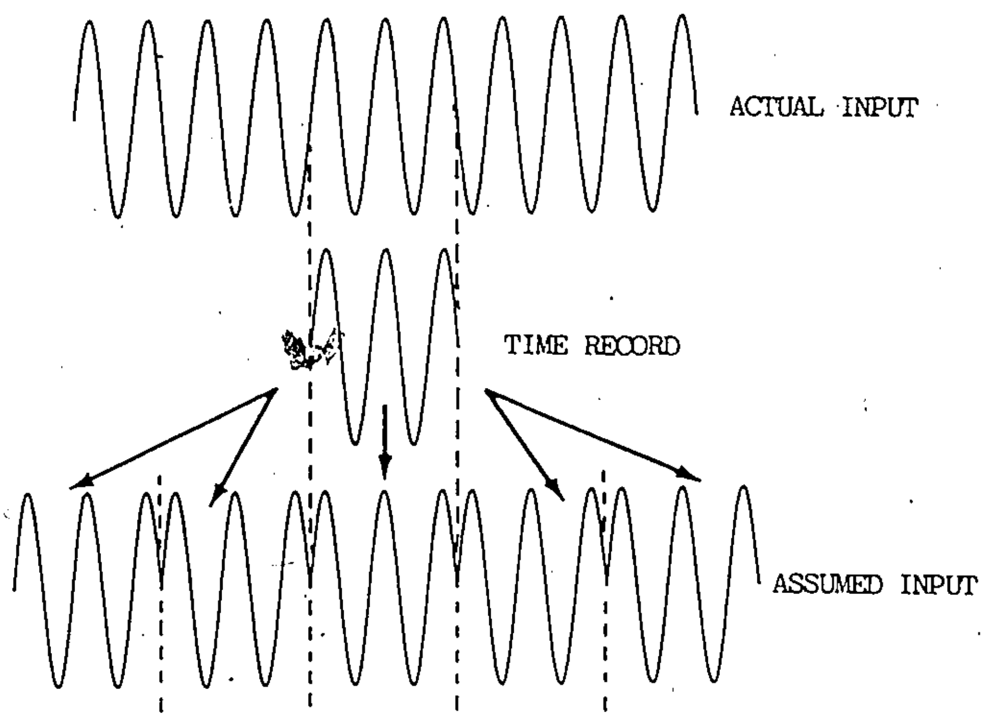


FIGURE 3.12: INPUT SIGNAL NOT PERIODIC IN TIME RECORD

that is equal to one of the 512 spectral lines.

If the input signal frequency falls between two spectral lines (see Figure 3.13) the peak cannot be observed because of the discreteness factor. Note that not only does the spectrum become broad, but the measured amplitude varies as a function of where the signal is in the spectrum.

3.1.5 Windowing

Windowing is a technique used in digital signal processing to reduce the errors associated with leakage. Figure 3.14 shows the effect windowing in the time domain and Figure 3.15a shows a sine wave not periodic in time record and its FFT results using no window and a flat top window is illustrated in Figure 3.15b. The flat-top window is used for increased amplitude resolution at the expense of frequency resolution. Other window functions include the "Hanning Window" which is generally used on broadband signals and for improved frequency resolution, and the "Rectangular Window" which is used for transient capture or signals periodic in the observation window.

3.1.6 Averaging

The spectrum generated from the analysis of one block of points is referred to as the instantaneous spectrum.

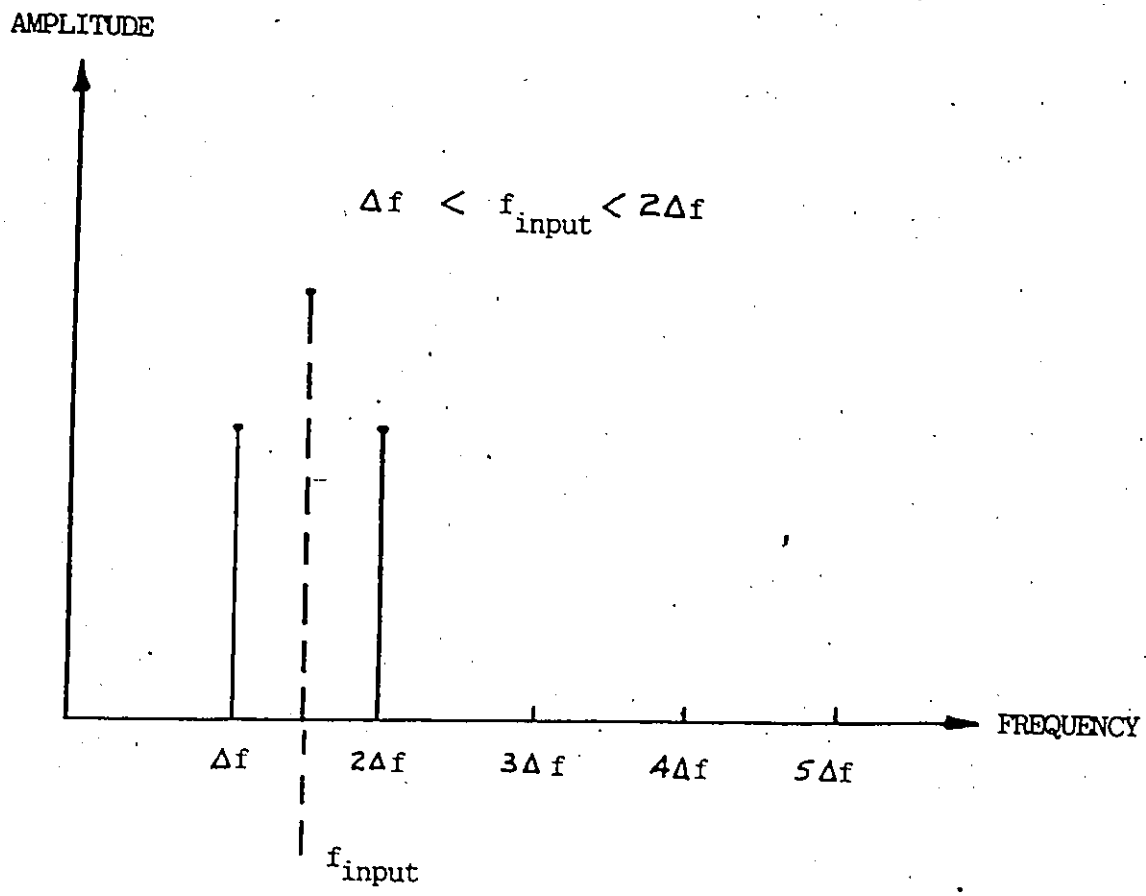


FIGURE 3.13: INPUT SIGNAL FREQUENCY FALLS BETWEEN TWO SPECTRAL LINES

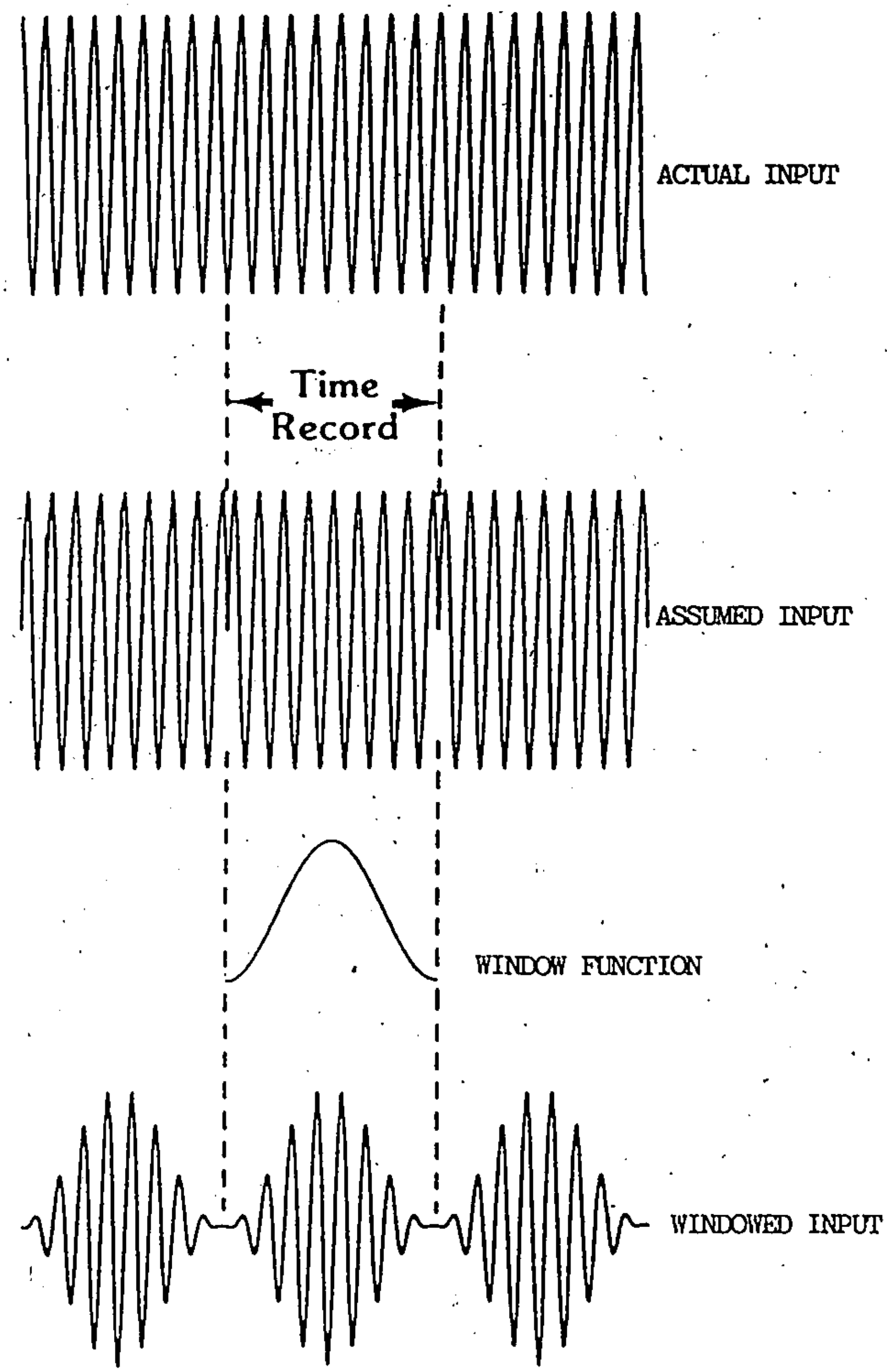


FIGURE 3.14: THE EFFECT OF WINDOWING IN THE TIME DOMAIN

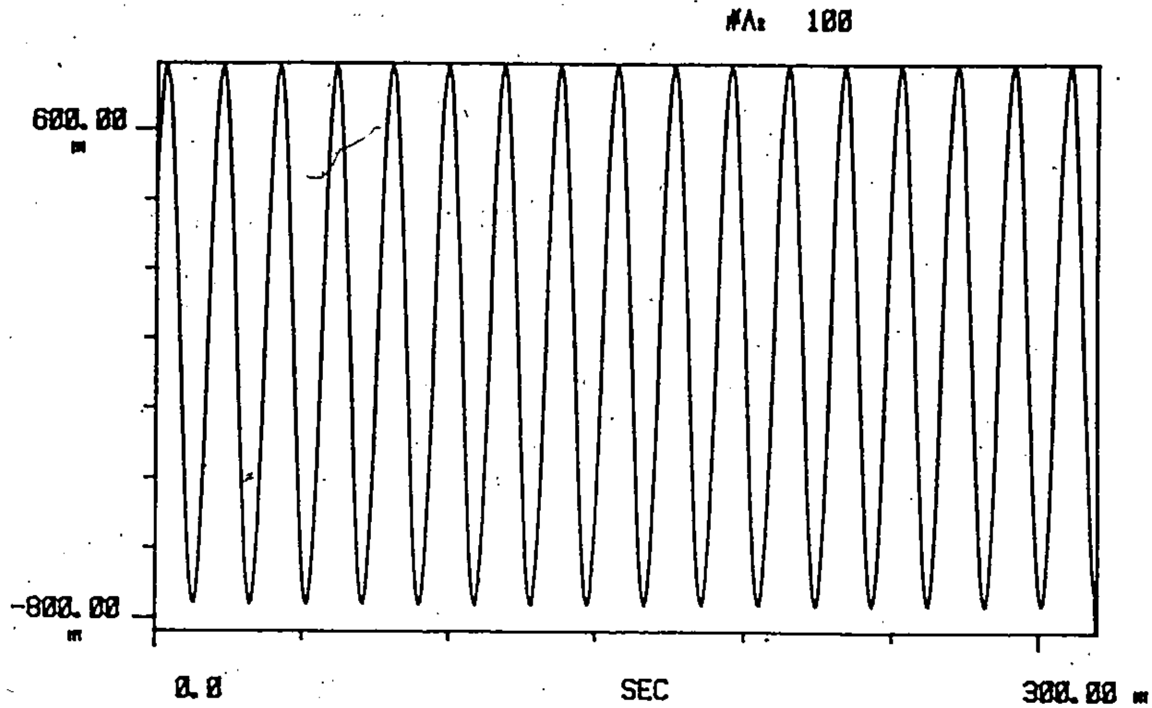


FIGURE 3.15a: SINE WAVE NOT PERIODIC IN TIME RECORD

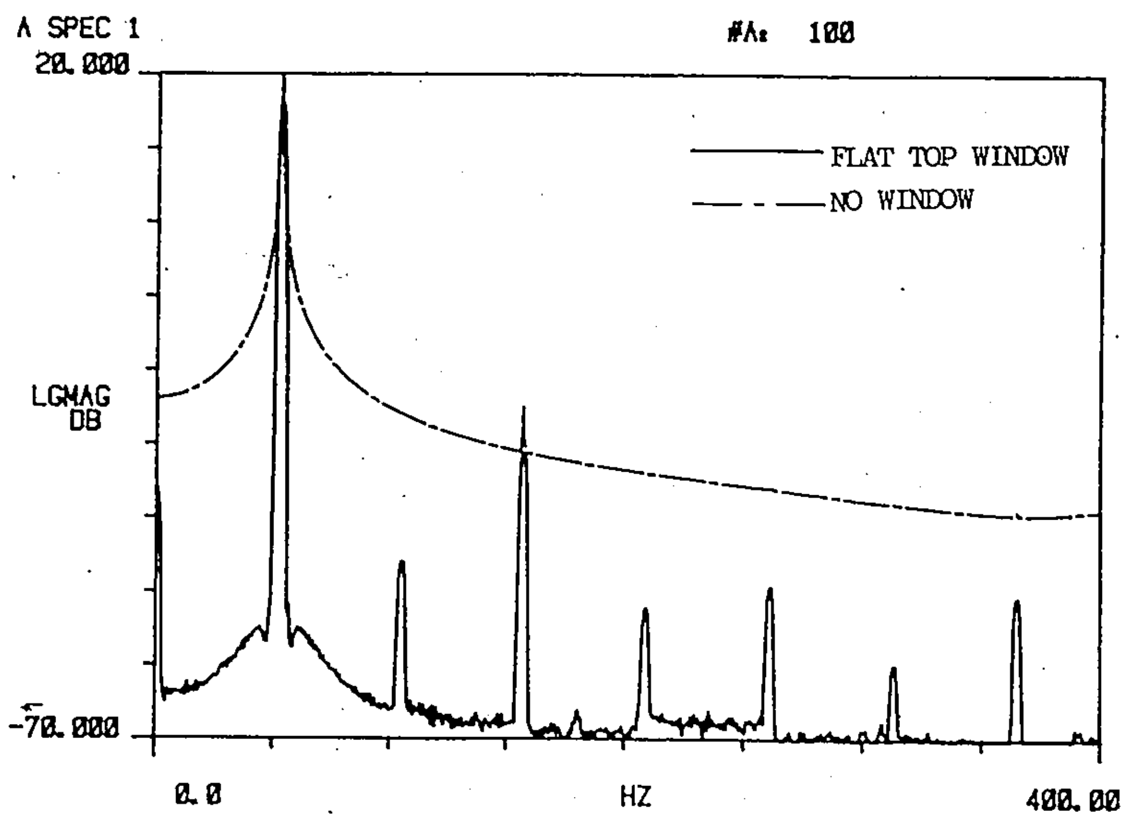


FIGURE 3.15b: FFT RESULTS OF 3.15a WITH NO WINDOWING AND WITH A FLAT-TOP WINDOW

As the input amplitude of the frequency changes, the instantaneous spectrum shows these changing effects almost immediately.

Many signals to be analyzed contain contamination such as noise, fluctuations in amplitude and so forth. A statistical mean or average of several "instantaneous" spectra will reduce the uncertainty and provide a cleaner representation of the spectrum. Therefore, many "instantaneous" spectra are averaged to yield one average spectrum.

There are three commonly used types of averaging:

a. Stable Averaging

Here, every individual spectrum is weighted equally and the average is always calibrated. Mathematically it is expressed as:

$$A_n = A_{n-1} + \frac{I_n - I_{n-1}}{n}$$

where:

A_n is the average after n time records,

A_{n-1} is the average after (n-1) time records,

I_n is the nth time record,

n is the number of time records.

b. Linear Averaging

Linear averaging is the sum of all individual spectrums divided by the total number of spectra. During the summation, the display of the summation is uncalibrated and continually grows. Mathematically it is expressed as:

$$A_n = \frac{\sum_{i=1}^{n} I_i}{n}$$

where:

A_n is the average after n time records

I_i is the i^{th} time record,

n is the number of time records.

c. Exponential Averaging

The most recent spectrum is more heavily weighted than previous spectra. Mathematically it is expressed as:

$$A_n = A_{n-1} + \frac{I_n - A_{n-1}}{N}$$

where:

A_n is the average after n time records,

A_{n-1} is the average after (n-1) time records,

I_n is the n^{th} time record,

N is the decay constant (in the HP5423A, it is equal to the number of averages selected).

3.2 Bearing Frequency Analysis

Many technical papers have been written on bearing frequency analysis. These papers usually present four rather complicated equations to calculate the frequencies of various bearing components in terms of the shaft speed. This section provides a clear development of these frequency equations.

Figure 3.16 illustrates a typical ball bearing. Normally, the bearing manufacturer will specify the following:

a. the number of balls,

b. the ball diameter,

c. the pitch diameter and

d. the contact angle.

Many bearings have a zero contact angle (thrust and deep grove bearings for example). However, a non-zero contact angle gives a radial bearing some axial thrust capability, so it is quite common to find a bearing with, say a 20 degree contact angle (such as the test bearing used in this experiment).

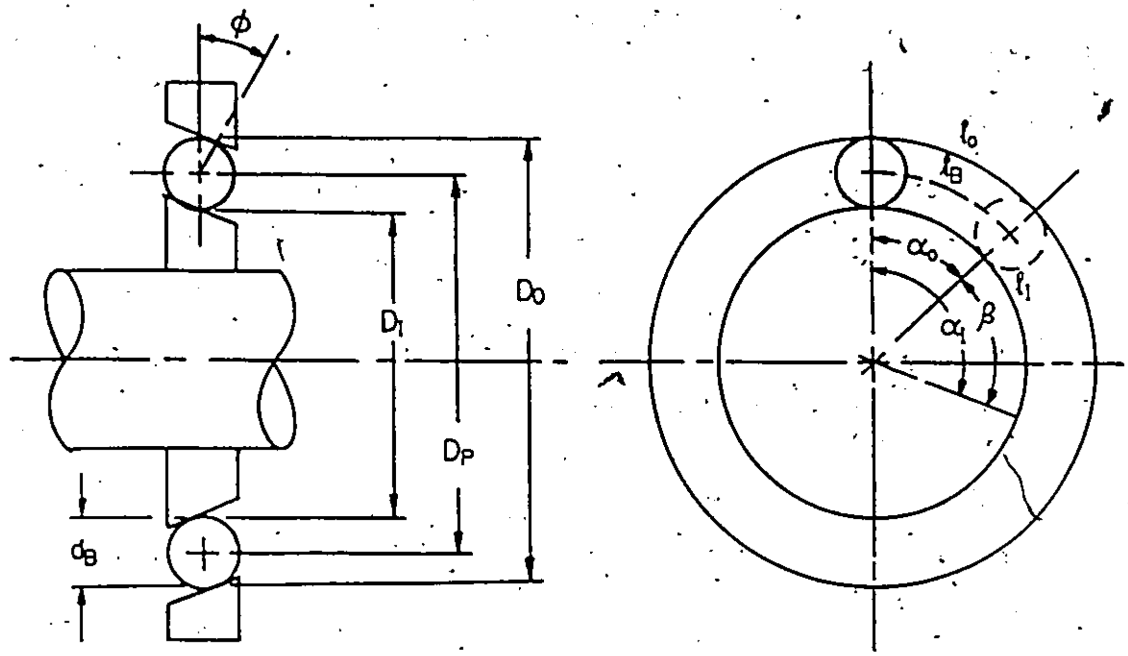


FIGURE 3.16: A TYPICAL BEARING GEOMETRY

With reference to Figure 3.16:

D_P is the pitch diameter,

d_B is the ball diameter,

N_B is the number of balls,

ϕ is the contact angle,

D_I is the inner race contact diameter = $D_P - d_B \cos \phi$,

D_O is the outer race contact diameter = $D_P + d_B \cos \phi$.

l_O is the linear travel of ball on outer race = πd_B

l_B is the linear travel of ball center (not equal to l_O)

l_I is the linear travel of inner race

α_O is the angular travel of ball center

α_I is the angular travel of inner race (and shaft)

β is the angular travel of inner race relative to ball

It is assumed that the outer race is locked into the bearing housing and cannot rotate (this is generally the case in most installations but, however, it is not essential to the analysis).

As seen from Figure 3.16, for one ball revolution, the ball has moved along the outer race one ball circumference and is at a new angular position. The inner race has moved one ball circumference relative to the ball center and is also at a new position.

$$\ell_o = \pi d_B \tag{3.2.1}$$

where ℓ_o is the linear travel of ball on the outer race. The ratio of this distance to the outer race circumference is the same as the ratio of the new angular position of the ball to a full circle of 2π radians.

Therefore, $$\frac{\alpha_o}{2\pi} = \frac{\ell_o}{\pi D_o} = \frac{\pi d_B}{\pi D_o}$$

$$\alpha_o = \frac{2\pi d_B}{D_o} \tag{3.2.2}$$

where α_o is the angular travel of ball center. The ratio of the distance travelled by the ball center to the pitch circumference (πD_p) is also the ratio of the new ball position to a full circle.

$$\frac{\ell_B}{\pi D_P} = \frac{\alpha_o}{2\pi} = \frac{2\pi d_B}{2\pi D_O}$$

$$\ell_B = \frac{\pi d_B D_P}{D_O} \qquad (3.2.3)$$

where ℓ_B is the linear travel of ball center.

The linear distance travelled by the inner race relative to the ball center is again the ball circumference (πd_B). The ratio of this distance to the inner circumference is the same as the ratio of the angular change in the inner race (relative to the ball) to a full circle.

$$\frac{\pi d_B}{\pi D_I} = \frac{\beta}{2\pi}$$

$$\beta = \frac{2\pi d_B}{D_I} \qquad (3.2.4)$$

But the ball has already moved through an angle of α_o, so the total angle change for the inner race is $\alpha_o + \beta$, where α_o is the angular travel of ball center.

$$\alpha_I = \alpha_o + \beta$$

where α_I is the angular travel of inner race (and shaft).

Therefore, $\alpha_I = \dfrac{2\pi d_B}{D_I} + \dfrac{2\pi d_B}{D_O}$

$$\alpha_I = 2\pi d_B \left[\frac{1}{D_I} + \frac{1}{D_O} \right] \qquad (3.2.5)$$

Finally, the total linear distance travelled by the inner race is found from the ratio of the total angular movement to a full circle.

$$\frac{\ell_I}{\pi D_I} = \frac{\alpha_I}{2\pi} = \frac{2\pi d_B}{2\pi}\left[\frac{1}{D_I} + \frac{1}{D_O}\right]$$

$$\ell_I = \pi d_B \left[1 + \frac{D_I}{D_O}\right] \qquad (3.2.6)$$

The ratio of the movement of the inner race to the outer race is obtained by dividing equation (3.2.6) by equation (3.2.1), i.e.,

$$\frac{\ell_I}{\ell_O} = 1 + \frac{D_I}{D_O}$$

For a ball bearing, this ratio is always less than 2 because the inner race diameter is always less than the outer race diameter.

Assuming that the ball is spinning at a frequency f_B, revolution per second, the linear velocity of the ball is then,

$$V_B = f_B \ell_B \qquad (3.2.7)$$

Similarly, the linear velocity of the inner race is,

$$V_I = f_B \ell_I \qquad (3.2.8)$$

This is the same velocity as the rotational shaft frequency f_R, times the inner race circumference.

Therefore, $f_B \ell_I = f_R \pi D_I$

$$f_B = \frac{f_R \pi D_I}{\ell_I} \tag{3.2.9}$$

Substituting D_I and equation (3.2.6) into equation (3.2.9), we obtain,

$$f_B = f_R \frac{\pi (D_P - d_B \cos \phi)}{\pi d_B (1 + D_I/D_O)}$$

$$f_B = \frac{f_R (D_P - d_B \cos \phi)}{d_B \left(\dfrac{D_P - d_B \cos \phi}{D_P + d_B \cos \phi} + 1 \right)}$$

hence,

$$f_B = f_R \frac{D_P}{2d_B} \left[1 - \left(\frac{d_B}{D_P} \cos \phi \right)^2 \right] \tag{3.2.10}$$

Thus the ball spin frequency is given by equation (3.2.10) in terms of rotational shaft frequency (rps), ball diameter, pitch diameter and contact angle.

The ball assembly or cage will rotate on the pitch diameter at the ball velocity V_B. The assembly frequency is therefore,

$$f_A = \frac{V_B}{\pi D_P} = \frac{f_B \ell_B}{\pi D_P}$$

$$= \frac{f_B \pi d_B D_P}{D_O \pi D_P} = \frac{f_B d_B}{D_O}$$

Combining the preceeding expression with equation (3.2.10) and the definition of D_O, we obtain,

$$f_A = \frac{f_R}{2}\left[1 - \frac{d_B}{D_P}\cos\phi\right] \qquad (3.2.11)$$

This is the fundamental train frequency which is the rotational frequency of the balls and cage assembly.

In a bearing with perfectly round and smooth surfaces, these frequencies would be detected only if one of these components had a serious imbalance. Since the components are generally quite small, we would probably only see the shaft unbalance. Unfortunately, no man-made surface is perfect, so we can expect to see frequencies related to these components, at low levels, even in a new high quality precision bearing.

A single outer race defect will result in the generation of the ball pass frequency of the outer race f_O, as every ball in the assembly passes over the defect each time the assembly makes one complete revolution. Thus,

$$f_O = N_B f_A = f_R \frac{N_B}{2}\left[1 - \frac{d_B}{D_P}\cos\phi\right] \quad (3.2.12)$$

A single inner race defect will be contacted by every ball in the assembly each time the assembly makes one revolution round the shaft. Since the shaft and the assembly rotate in the same direction, the assembly rotates around the shaft at a frequency equal to the difference between the shaft and the assembly frequencies. Thus the ball pass frequency of the inner race f_I is equal to the number of balls multiplied by the relative rotational frequency.

$$f_I = N_B \left(f_R - f_A\right)$$

Substituting equation (3.2.11) into the above equation, we obtain,

$$f_I = \frac{N_B}{2} f_R \left[1 + \frac{d_B}{D_P} \cos \phi\right] \qquad (3.2.13)$$

A single ball defect will contact both inner and outer races and possibly the cage, each ball revolution. However, since a ball can spin in all directions, it is possible that a ball can miss one of the races, occasionally, giving erratic results. In general, a ball defect can be expected at twice the ball spin frequency.

In practice, a bearing will generally exhibit every possible sum and difference among the five frequencies, f_R, f_A, f_B, f_O and f_I. Multiple defects, overall roughness, contaminated lubricant and structural resonances may all combine to make the interpretation of a bearing vibration signature virtually impossible. However, the five main frequencies and their sums and differences will usually be present and in many cases a fault can be identified early enough to prevent major damage.

IV. IN-PLANT VIBRATION MONITORING EXPERIENCE

4.1 Introduction

In this section, results of a study of in-plant vibration monitoring at the Ford Motor Co. of Canada Ltd. are presented.

The main objectives of this study were:

a) To determine the feasibility of applying vibration monitoring techniques to high volume multi-station transfer machines employed at the Ford Essex engine plant.

b) To obtain representative overall vibration levels as well as frequency spectra from designated operations for use as "baseline" and "failure" data in a possible future vibration monitoring system.

The proposed study was to extend approximately six months to assure sufficient data for evaluation. However, during the first three months of monitoring, a number of failures were encountered. It was then decided that sufficient information was available to meet the objectives mentioned earlier.

4.2 Instrumentation

The following instruments were used in carrying the vibration monitoring process:

a) Bruel and Kjaer 2209, Precision Sound Level Meter,

b) Bruel and Kjaer 4291, Accelerometer Calibrator,

c) Bruel and Kjaer 4370, Accelerometer,

d) Hewlett Packard 5423A, Structural Dynamics Analyzer,

e) Hewlett Packard 9872B, Digital Plotter,

f) Teac R-61D, Tape Recorder.

Since some of these instruments were also used in other sections, their general descriptions were listed in Appendix C.

The schematic of the equipment setup for in-plant vibration measurements and laboratory analysis, is illustrated in Figure 4.1.

4.3 Measurement Methodology

The machining stations selected for vibration measurements were those identified by the plant maintenance personnel as being critical to the process or as having a history of requiring excessive maintenance. The machining operations and stations which were actually

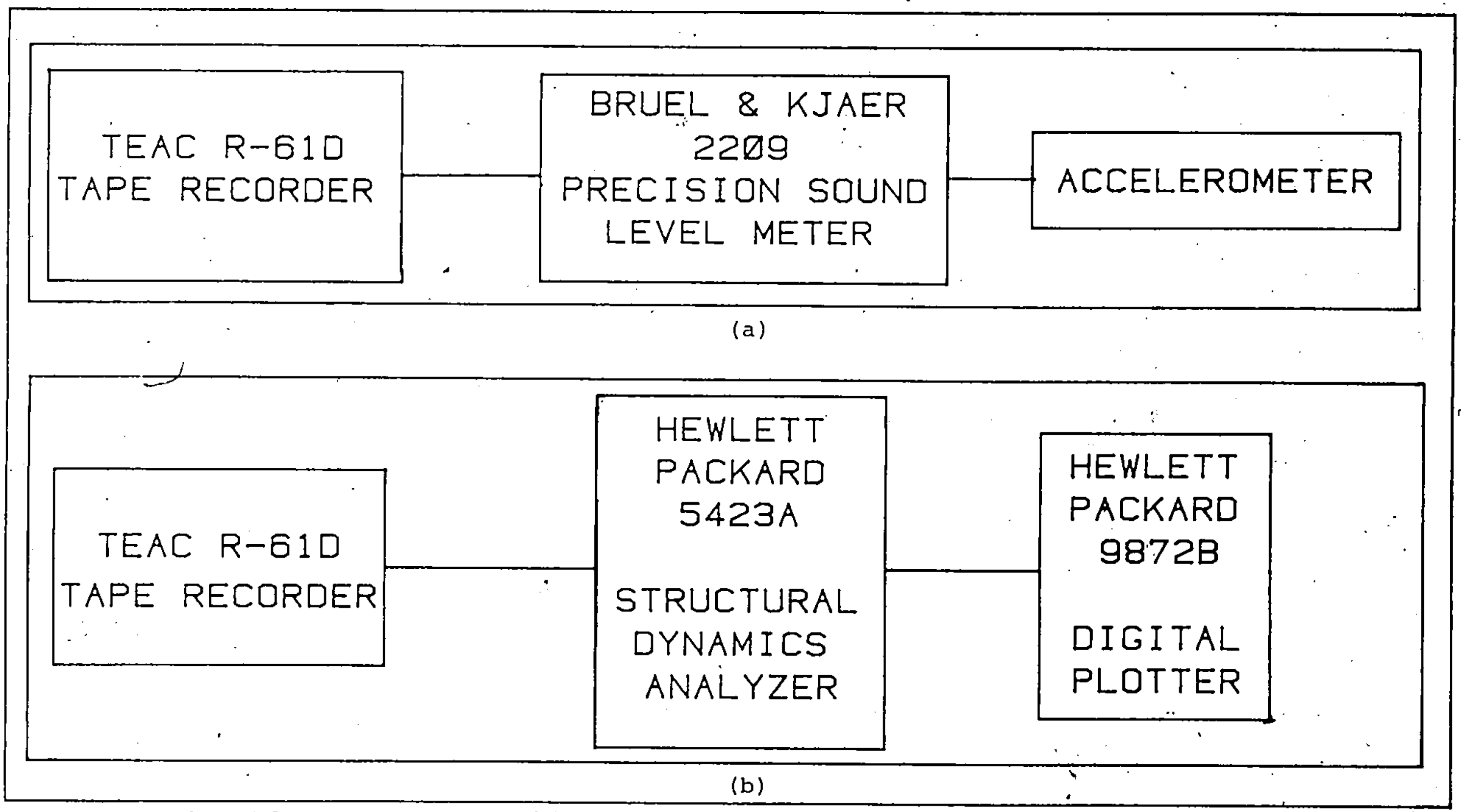


FIGURE 4.1: EQUIPMENT SET UP (a) IN-PLANT MEASUREMENTS (b) IN LABORATORY ANALYSIS

monitored, are summarized in Table 4.1.

A single accelerometer was used to measure accelerations at various measuring positions in each machining station. The accelerometer was mounted on a magnetic base for fast mounting and removal, thus minimizing disruptions of the transfer machine operations. The output signal from the accelerometer was conditioned in the sound level meter and then recorded on the cassette recorder on both F.M. and D.R. channels. This provided a system frequency response from 2 Hz to 8000 Hz. The recorded accelerometer signal was then analyzed in the laboratory using the structural dynamics analyzer (FFT analyzer).

All frequency spectra were averaged to minimize the effects of noise and random components. The number of averages varied with the measuring position as a direct function of the amount of time during which a useable signal was generated in a given process cycle.

The first week's data consisted of the "idle" measurements only i.e., the machines were rotating without machining. These measurements were made at a large number of positions at each machining station. This data was reviewed to determine the "best" measuring positions at each station based on signal output level and uniqueness of vibration spectrum. On this basis, the

DEPARTMENT	OPERATION	STATION	NUMBER OF MEASURING POSITIONS
Cylinder Head	20D South	14R	3
	20D South	12R	1
	20D North	14R	3
Crankshaft	90 South	35-36L	2
Camshaft	60 South	17R	3
	60 South	13R	3
Piston	50A	6R	4
Connecting Rod	60 North	3R	6
Cylinder Block	150 North	10	2
	10 South	10R	4
	140 South	10R	1
	160 South	3L	1
			33

TABLE 4.1: LIST OF OPERATIONS AND STATIONS MONITORED

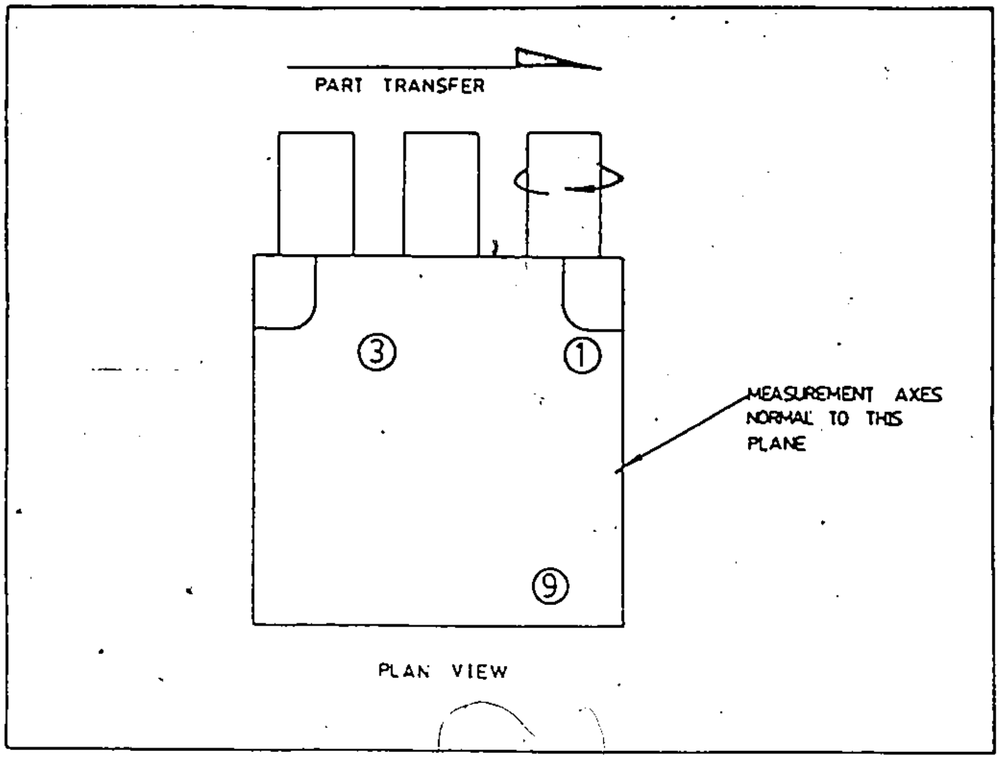
final measuring positions for each machining station were determined and are illustrated in Figures 4.2a to 4.9a. After the first week, a complete machining cycle was recorded at each position so as to provide both an "idle" and "machining" measurements. On certain occasions, only idle measurement could be obtained and, once again, this is apparent in the data presented.

Generally, measurements were taken weekly at the designated positions. However, on some occasions, a particular operation would not be in production hence no measurement would be taken at all due to scheduling conflicts. These omissions will become obvious as the data is presented.

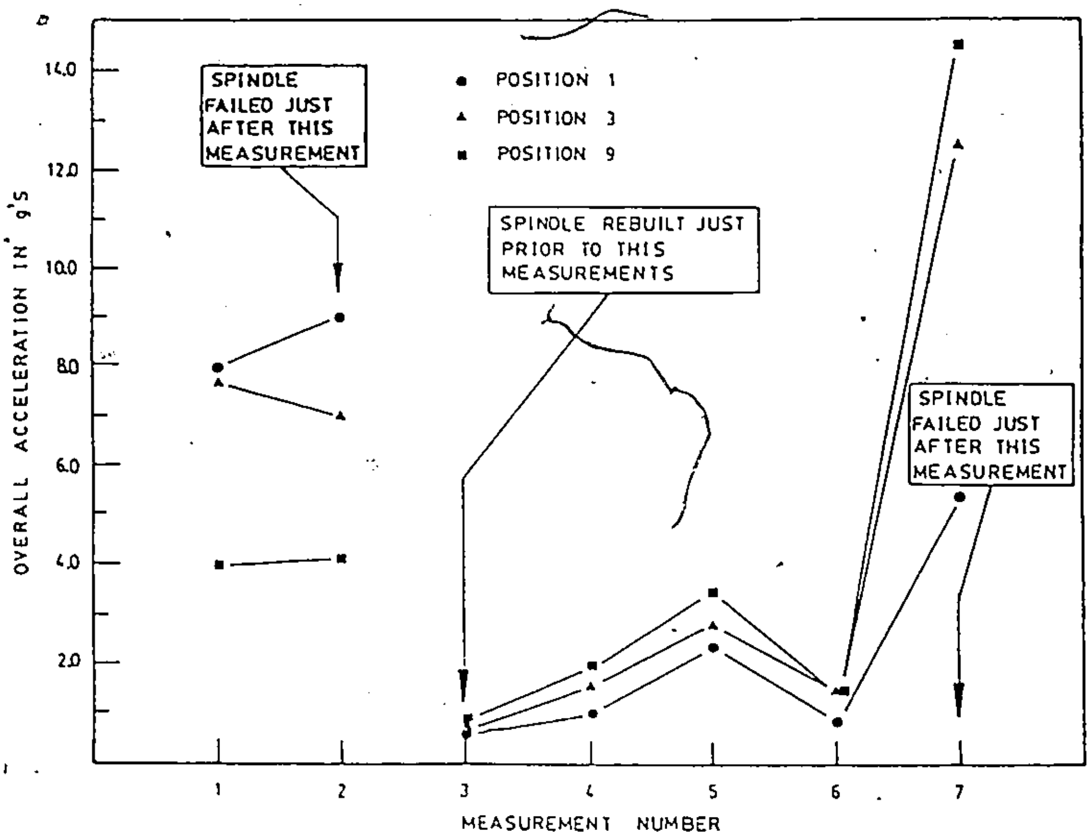
The use of a sound level meter coupled with an accelerometer, permitted the determination of the overall root-mean-square acceleration level at each measuring position. The overall acceleration levels for each measuring position are summarized in Table 4.2. The data is presented in graphical form in Figures 4.2b to 4.9b.

4.4 Results and Discussions

To determine if vibration monitoring was indeed feasible for use on the transfer machines, it was necessary to ensure that repeatable vibration measurements could be made at predetermined monitoring positions and also, if



(a)



(b)

FIGURE 4.2: CYLINDER HEAD, OPERATION 20D NORTH STA. 14 R.
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

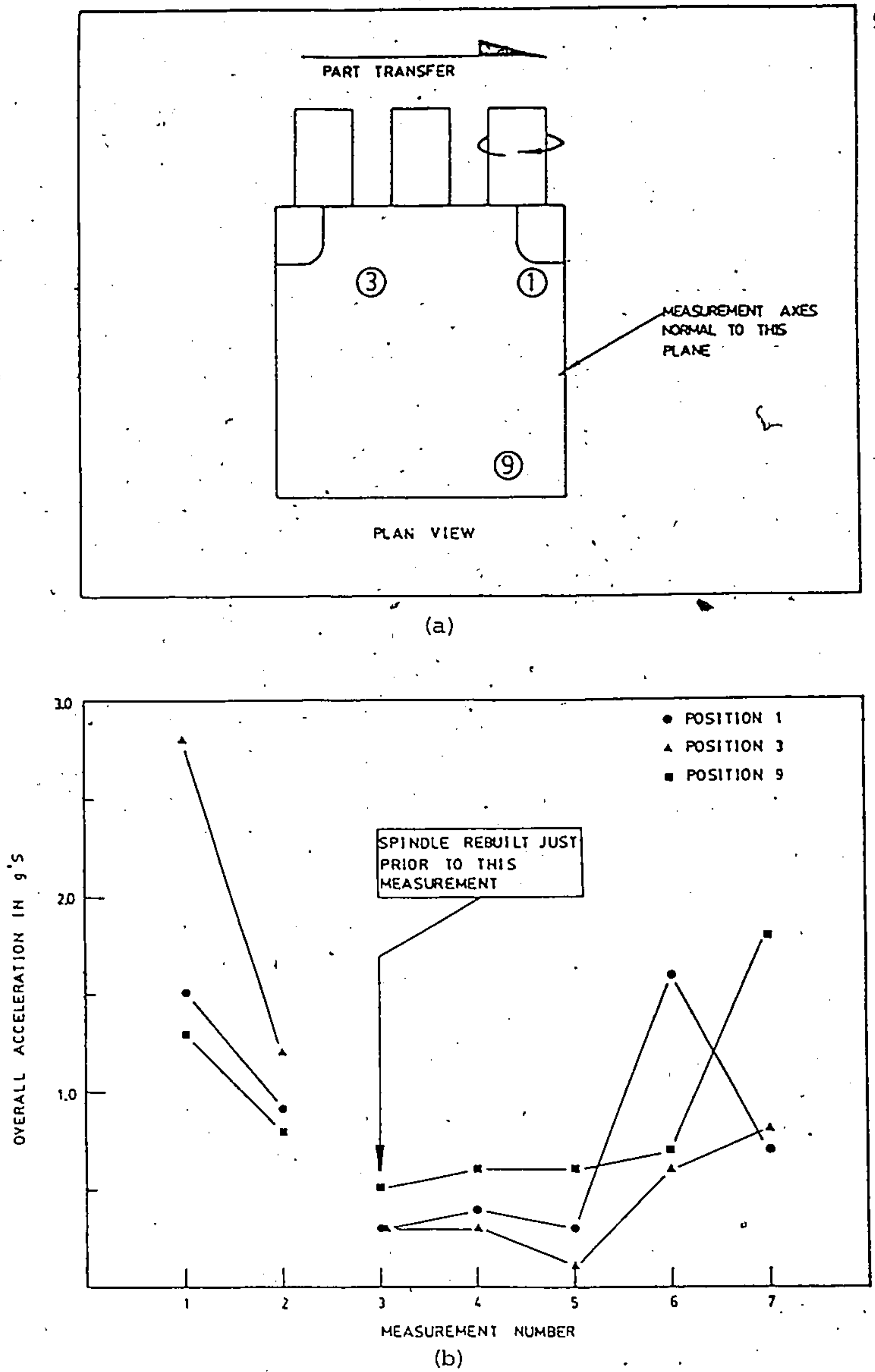
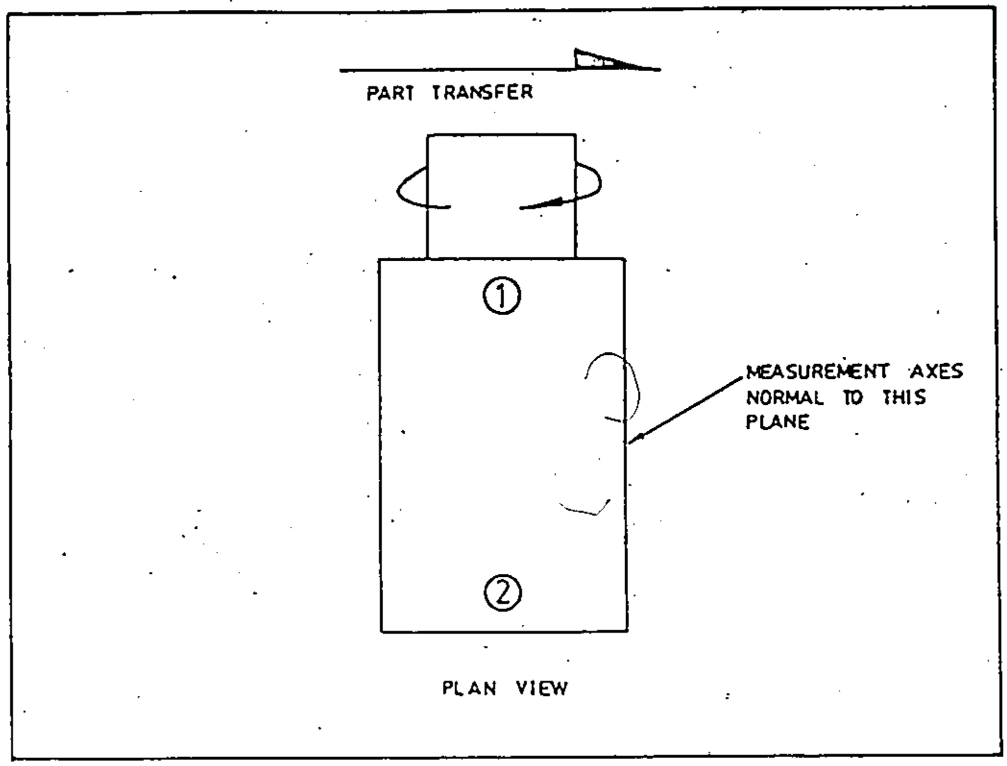
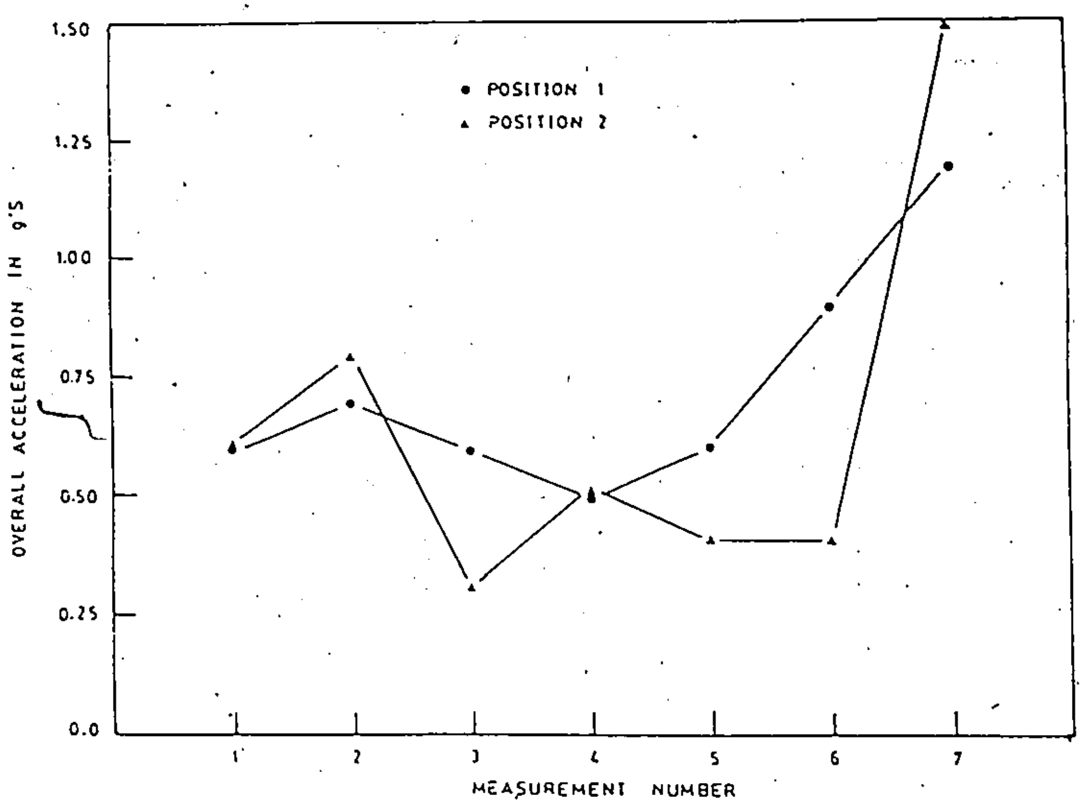


FIGURE 4.3: CYLINDER HEAD, OPERATION 20D SOUTH STA. 14R
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

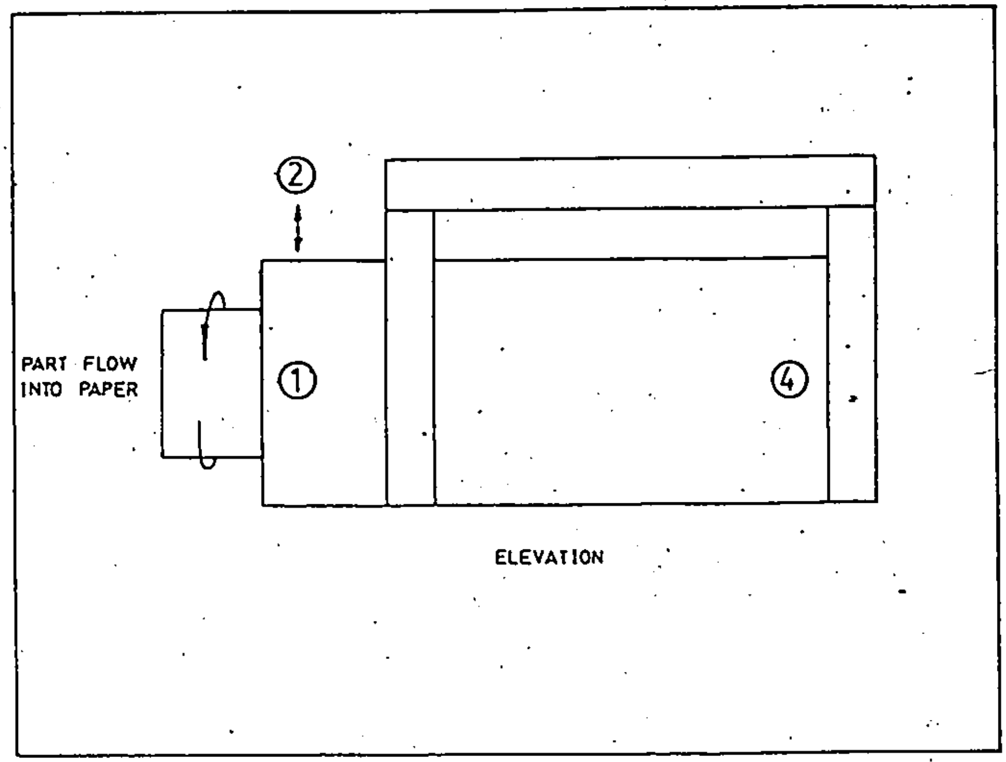


(a)

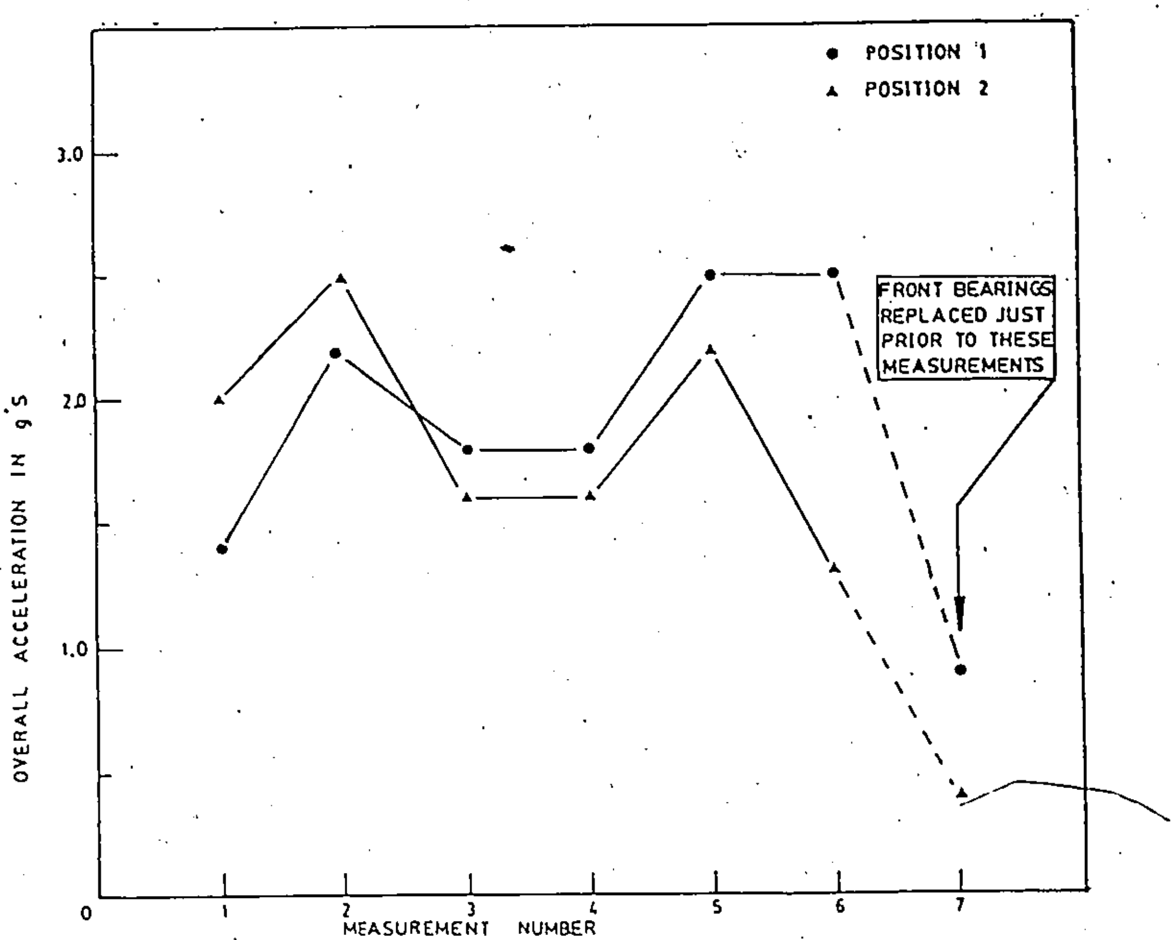


(b)

FIGURE 4.4: CRANKSHAFT, OPERATION 90 SOUTH STA. 35-36 L
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

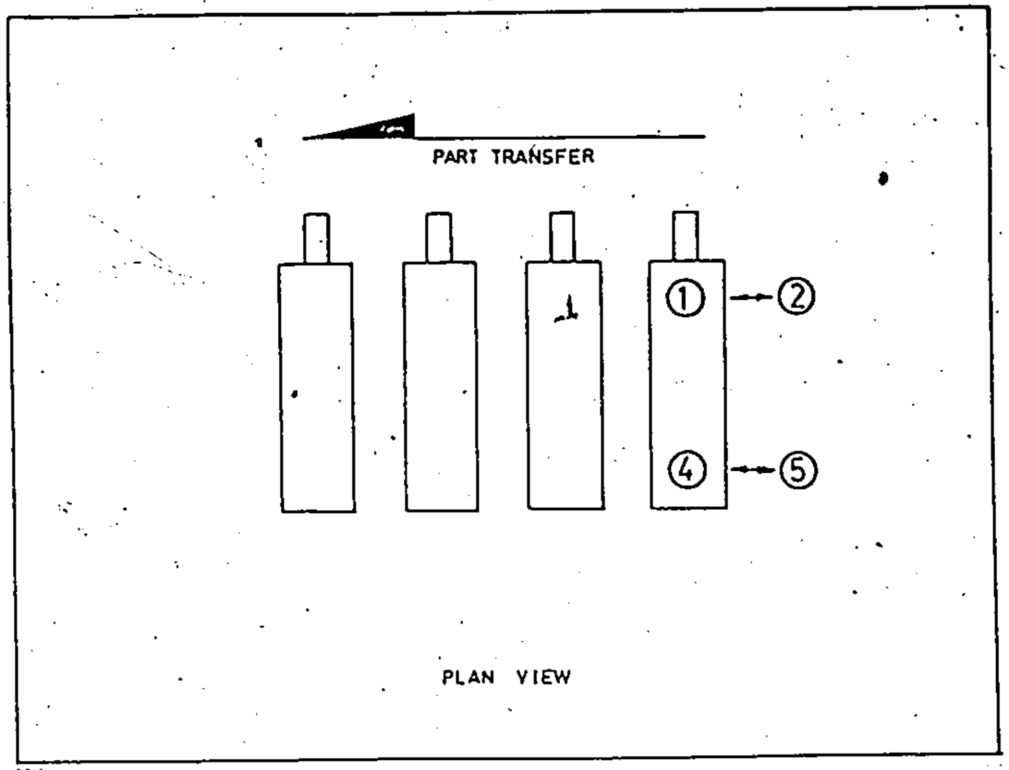


(a)

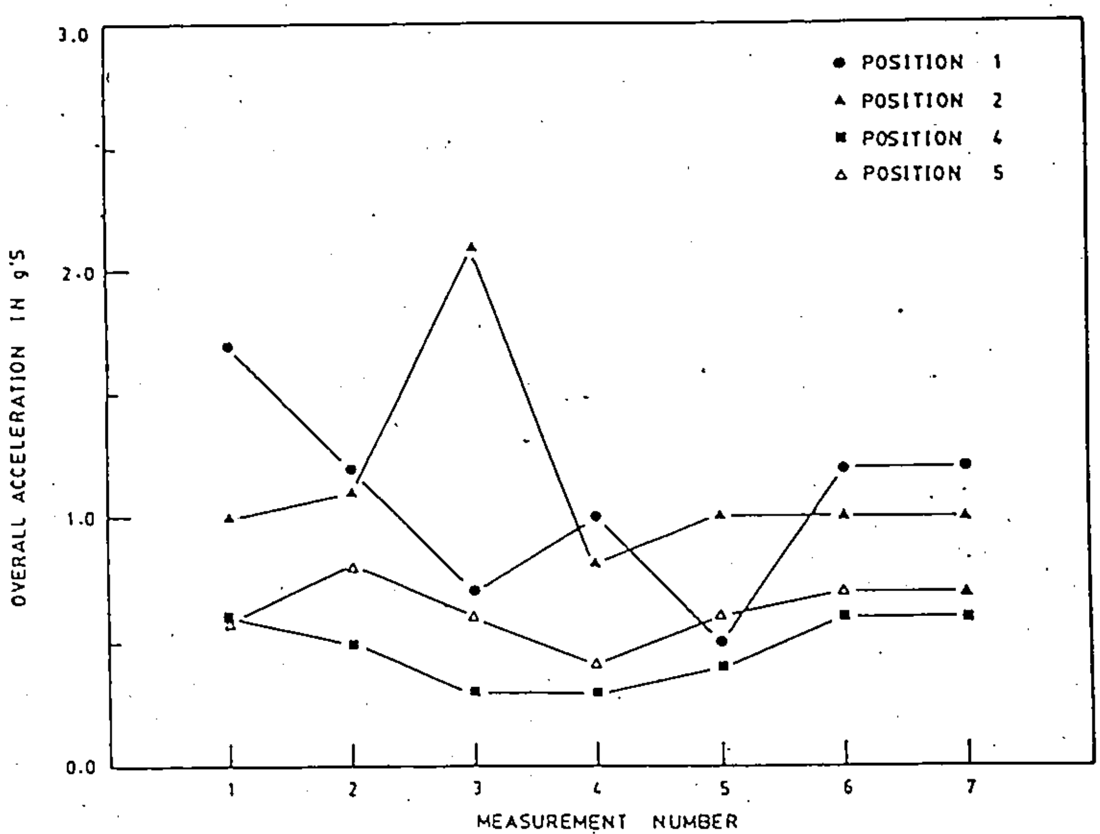


(b)

FIGURE 4.5: CAMSHAFT, OPERATION 60 SOUTH STA. 17R
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

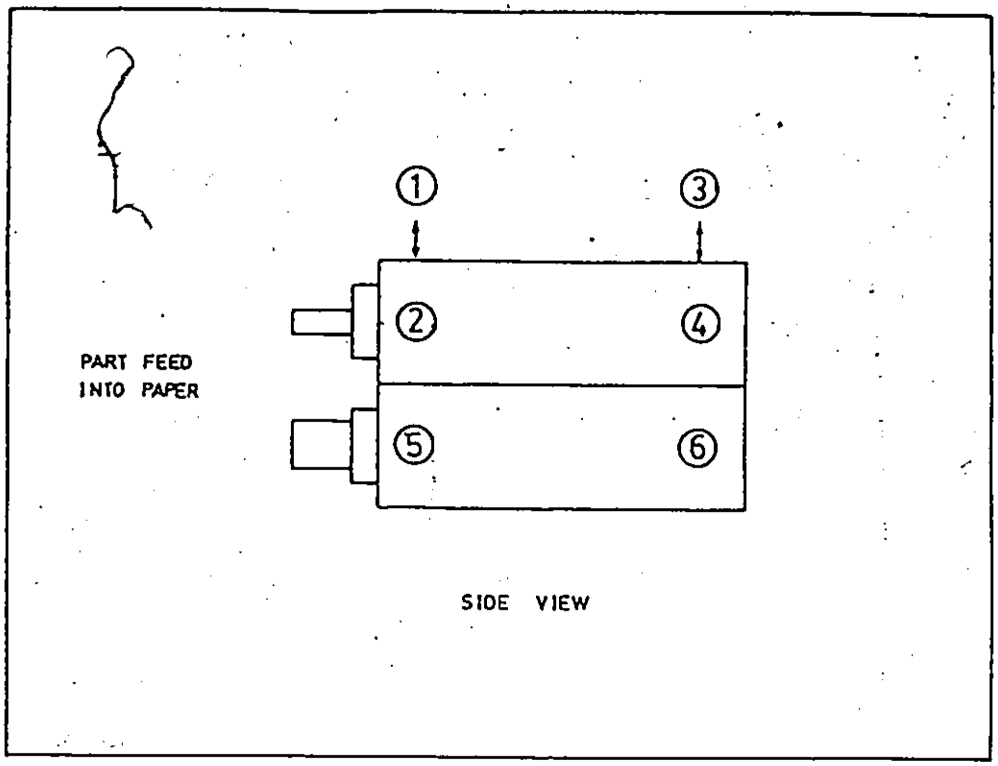


(a)

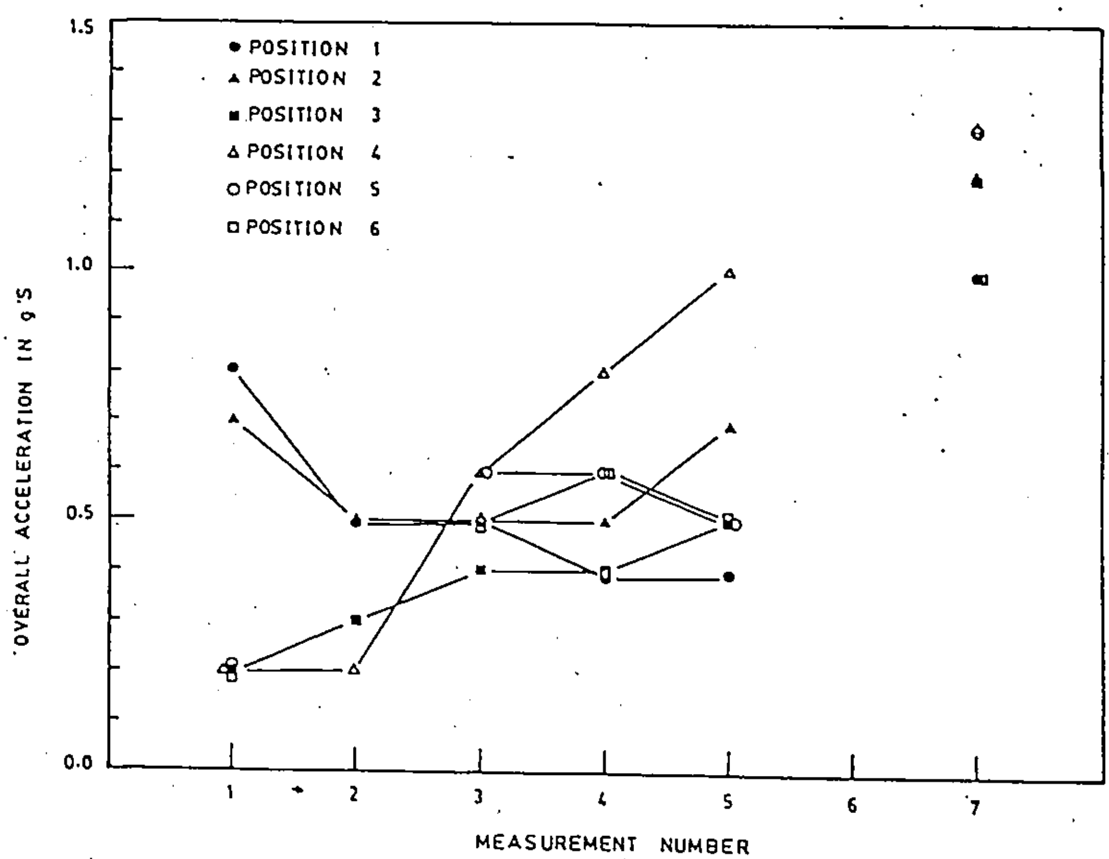


(b)

FIGURE 4.6: PISTON, OPERATION 50A STA. 6R.
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

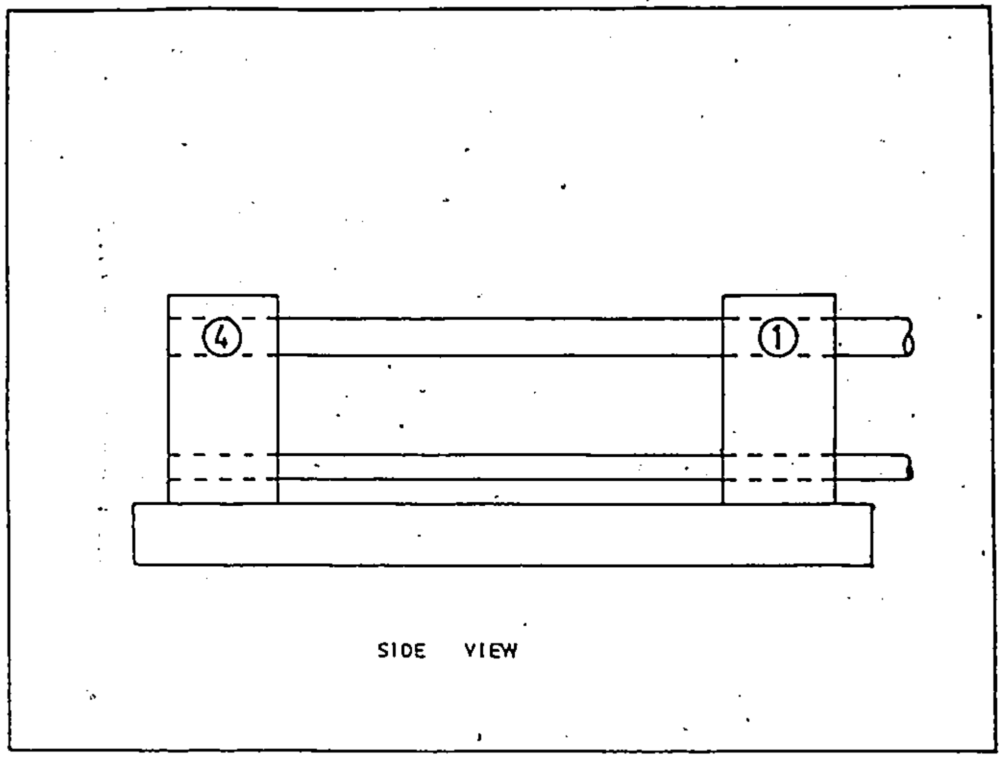


(a)

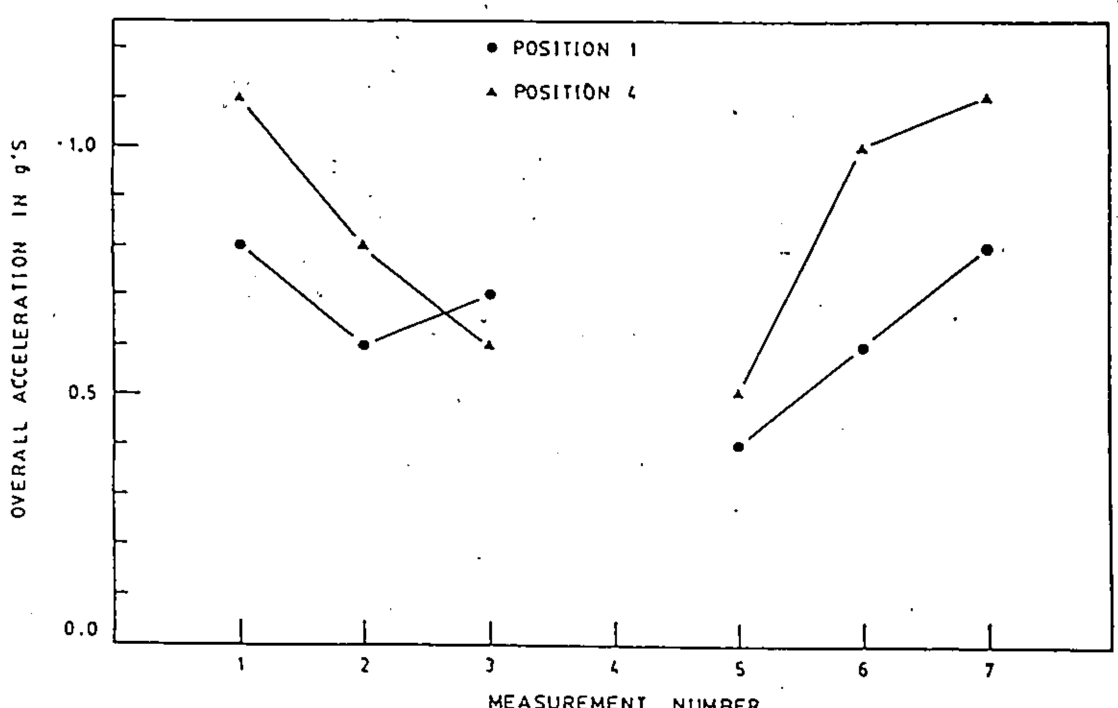


(b)

FIGURE 4.7: CONNECTING ROD, OPERATION 60 NORTH STA. 3R.
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

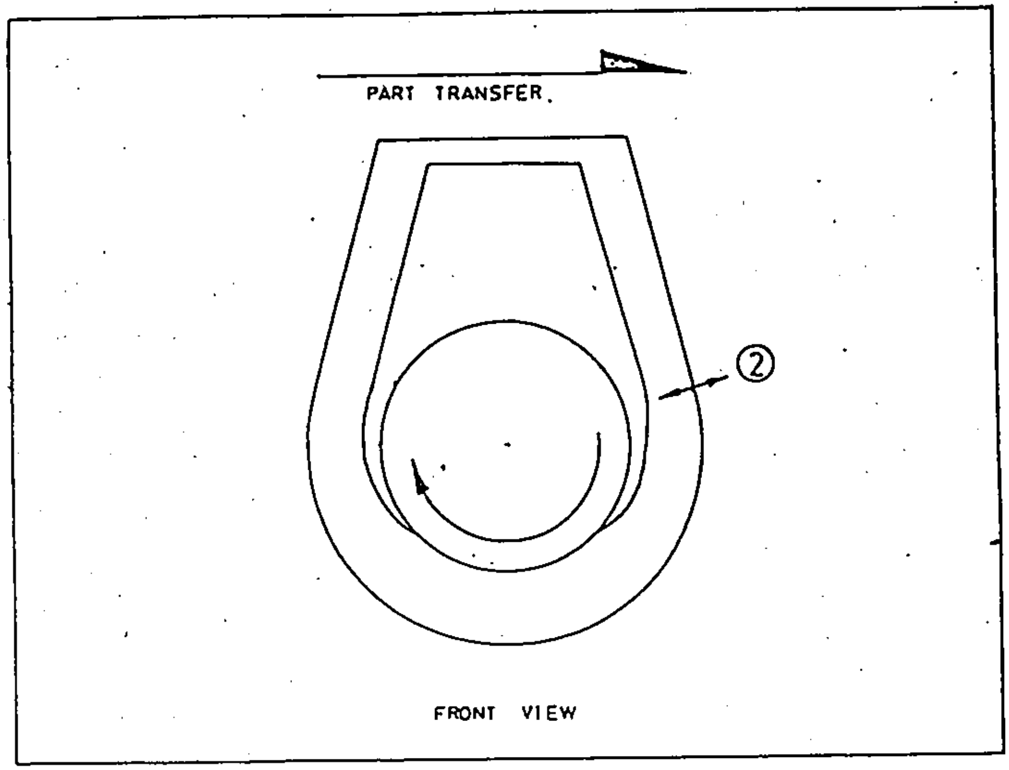


(a)

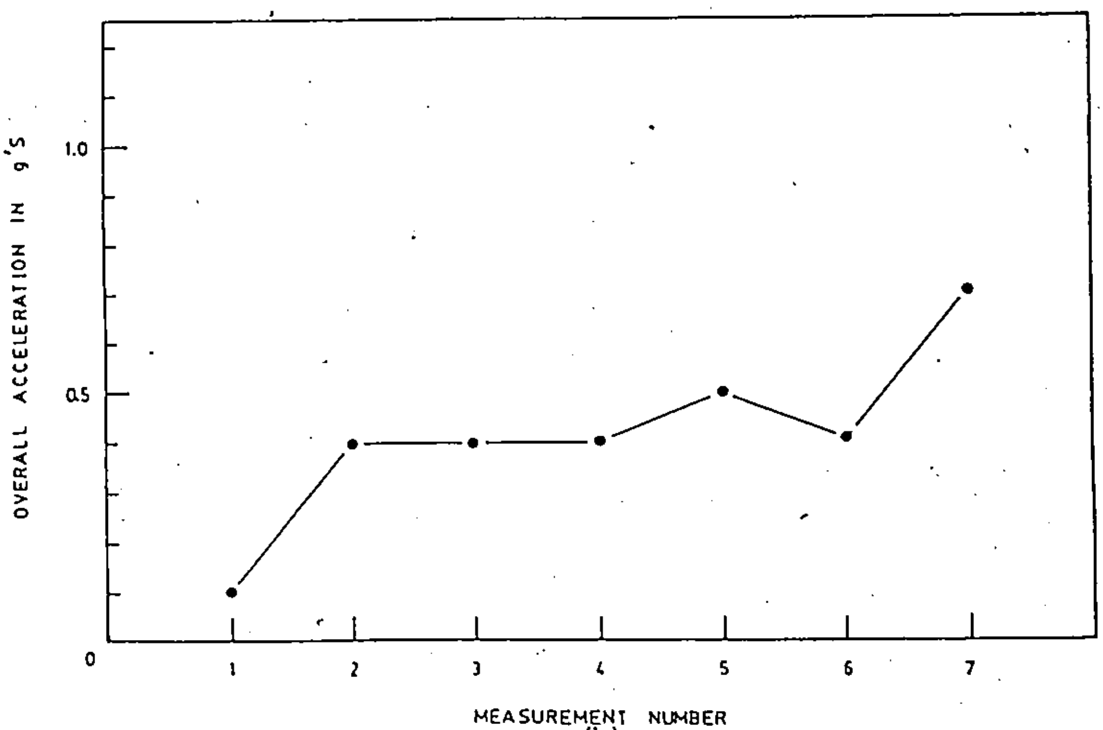


(b)

FIGURE 4.8: CYLINDER BLOCK, OPERATION 150 NORTH STA. 10
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER



(a)



(b)

FIGURE 4.9: CYLINDER BLOCK, OPERATION 160 SOUTH STA. 3 L.
(a) MEASURING POSITIONS
(b) OVERALL ACCELERATION VERSUS MEASUREMENT NUMBER

DATE	Cylinder Head							Crankshaft		Camshaft						Piston				Con. Rod						Cylinder Block							
	Op. 20D South				Op. 20D North			Op. 90 South		Op. 60 South						Op. 50A				Op. 60 North						Op. 10 South				Op.140 South	Op.150 North		Op.160 South
	Sta.14R			Sta.12R	Sta.14R			Sta.35-36L		Sta.17R			Sta.13R			Sta.6R				Sta.3R						Sta.10R				Sta.10R	Sta.10		Sta.3L
	1	3	9	3	1	3	9	1	2	1	2	4	1	2	4	1	2	4	5	1	2	3	4	5	6	1	2	4	5	1	1	4	2
June 28	1.5*	2.8*	1.3*	-	8.0*	7.7*	4.0*	0.6*	0.6*	1.4*	2.0*	0.8*	0.2*	0.2*	0.1*	1.7*	1.0*	0.6*	0.6*	.8*	.7*	.2*	.2*	.2*	.2*	0.3*	0.2*	0.2*	0.2*	-	0.8*	1.1*	0.1*
July 20	0.9	1.2	0.8	-	9.0	7.0	4.1	0.7	0.8	2.2	2.5	1.3	0.2	0.3	0.6	1.2	1.1	0.5	0.8	.5	.5	.3	.2	-	-	0.1*	0.1*	0.2*	0.2*	0.3	0.6	0.8	0.4
July 28	0.3	0.3	0.5	-	0.6	0.7	1.0	0.6*	0.3*	1.8	1.6	0.8	0.2	0.2	0.1	0.7	2.1	0.3	0.6	.5	.5	.4	6	6	.5	-	-	2.0	2.5	0.3	0.7	0.6	0.4
Aug. 4	0.4	0.3	0.6	5.0	1.1	1.7	2.0	0.5*	0.5*	1.8	1.6	0.8	-	-	-	1.0	0.8	0.3	0.4	.4	.5	.4	.8	6	.6	-	-	-	-	0.3	-	-	0.4
Aug. 11	0.3	0.1	0.6	1.2	1.4	2.8	3.5	0.6*	0.4*	2.5	2.2	-	-	-	-	0.5	1.0	0.4	0.6	.4	.7	.5	1.0	.5	.5	2.3	2.3	0.5	-	-	0.4	0.5	0.5
Aug. 25	1.6	0.6	0.7	2.0	0.9	1.6	1.6	0.8*	0.4*	2.5	1.3	1.5	-	-	-	1.2	1.0	0.6	0.7	-	-	-	-	-	-	-	-	-	-	0.2*	0.6	1.0	0.4
Sept. 8	0.7	0.8	1.8	3.4	5.4	12.5	14.5	1.2*	1.5*	0.9	0.4	0.4	-	-	-	-	-	-	-	1.0	1.2	1.2	1.3	1.3	1.0	6.1	8.0	4.5	5.5	0.5	1.1	0.8	0.7

TABLE 4.2: SUMMARY OF OVERALL VIBRATION LEVELS

* IDLE MEASUREMENT

distinct changes develop, both in the overall vibration level as well as spectral amplitudes, as a result of component failure.

The results shown in Figures 4.6b, 4.7b and 4.9b, and their corresponding spectra for one selected measuring position on the machining station, Figures 4.10 to 4.12, indicate that the processes were operating normally with no evident component failures and that their overall acceleration levels and spectra showed no significant changes over the study period. In fact, some slight fluctuations were evident. This was most likely due to the use of a magnetic mount for the accelerometer since there was no practical way of ensuring exactly the same holding force from measurement to measurement nor exact relocation of the accelerometer. Permanent mounting of the accelerometer would remove this particular difficulty.

The discussions in this section are limited to representative spectra only.

Figure 4.2 illustrates measuring positions and overall vibration data for operation 20D North, Station 14R, Cylinder Head. Its corresponding spectra are shown in Figure 4.13. These Figures clearly show the development of component failure, i.e., the changing overall vibration levels and spectra as the component failure progresses (in this case a bearing). Figure 4.13 documents the spectra associated with both "as new" and "failed"

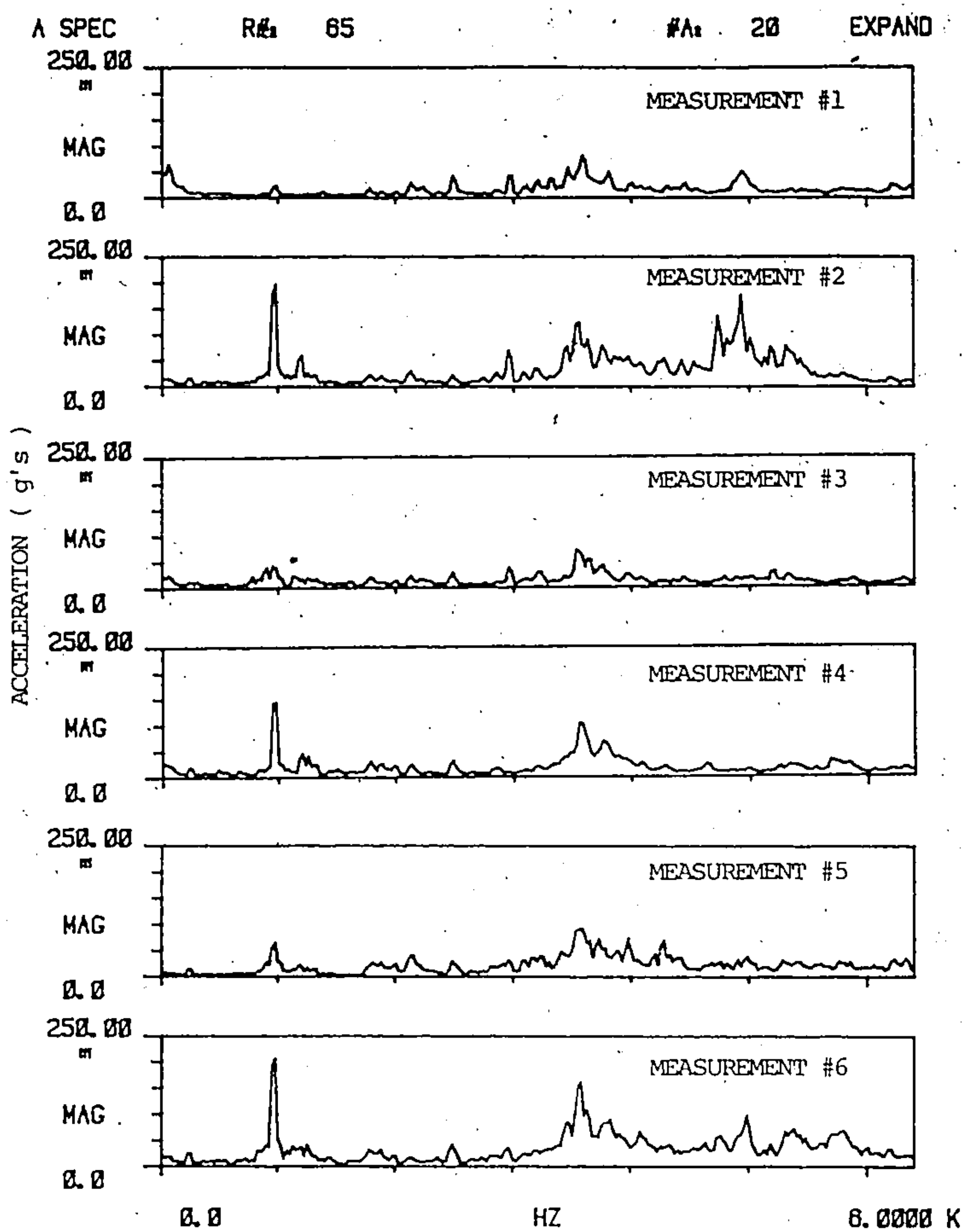


FIGURE 4.10: SPECTRAL DATA FOR PISTON OPERATION 50A STA. 6R. (MEASURING POSITION 1)

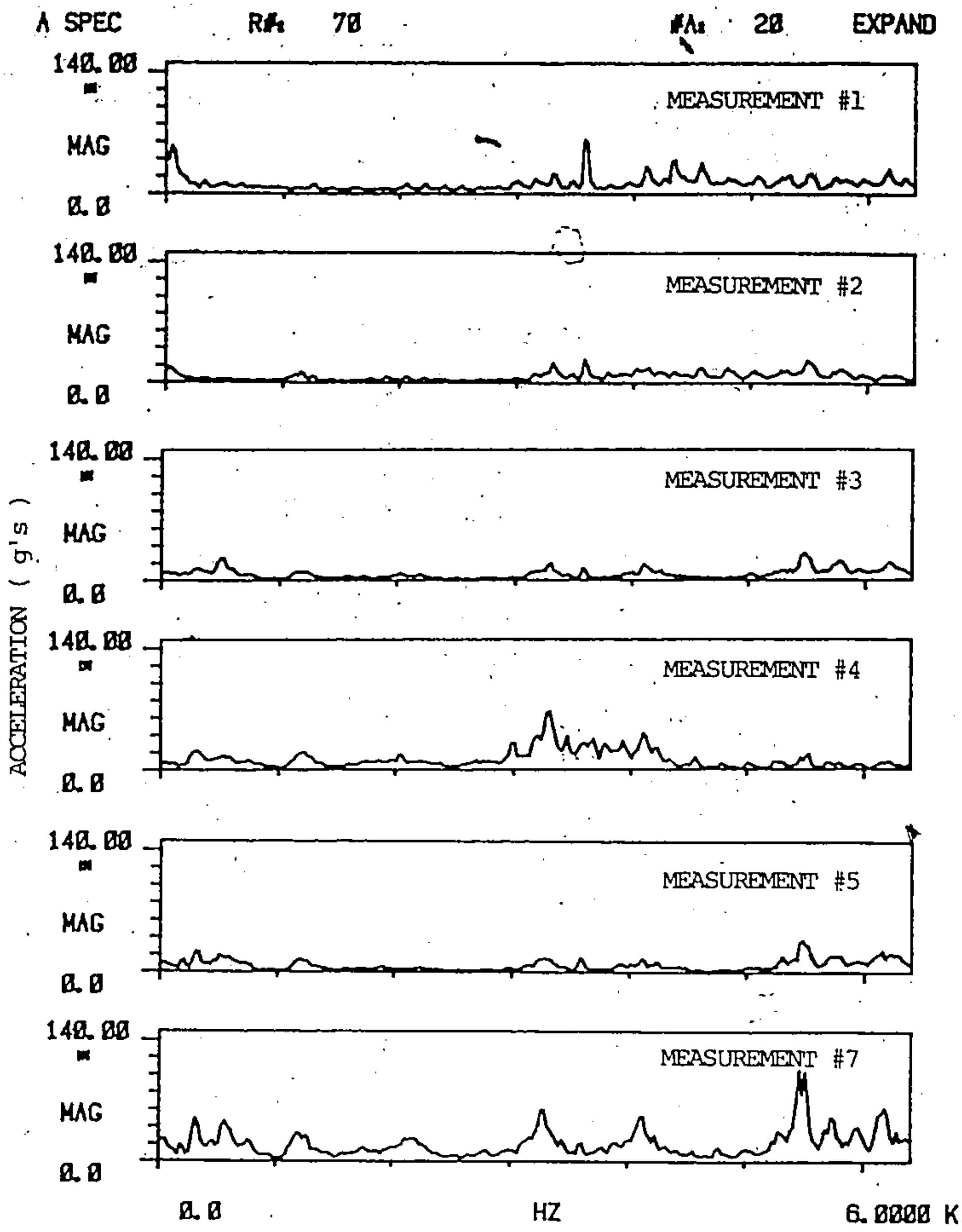


FIGURE 4.11: SPECTRAL DATA FOR CONNECTING ROD OPERATION 60 NORTH STA. 3R. (MEASURING POSITION 2)

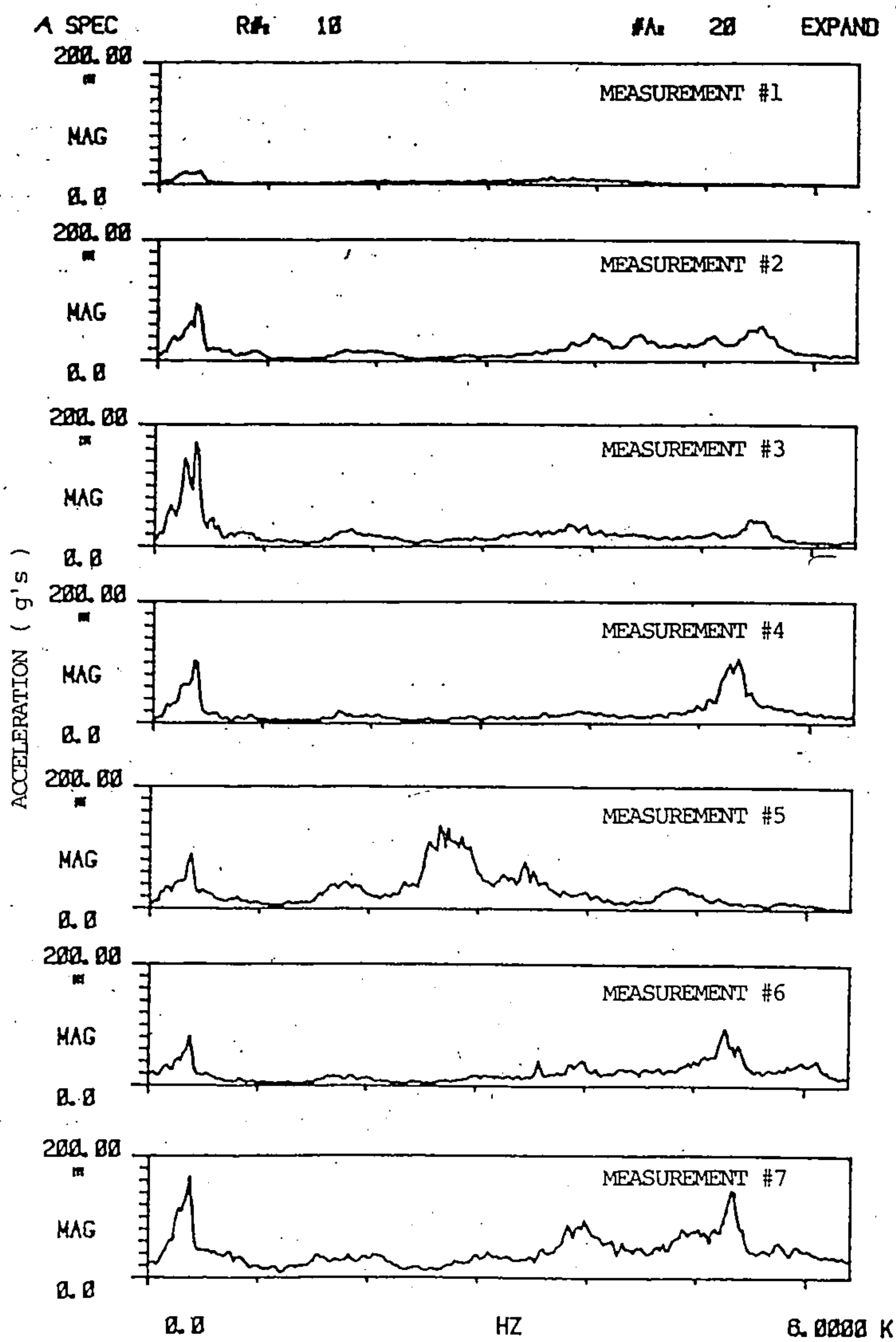


FIGURE 4.12: SPECTRAL DATA FOR CYLINDER BLOCK OPERATION 160 SOUTH STA. 3L. (MEASURING POSITION 2)

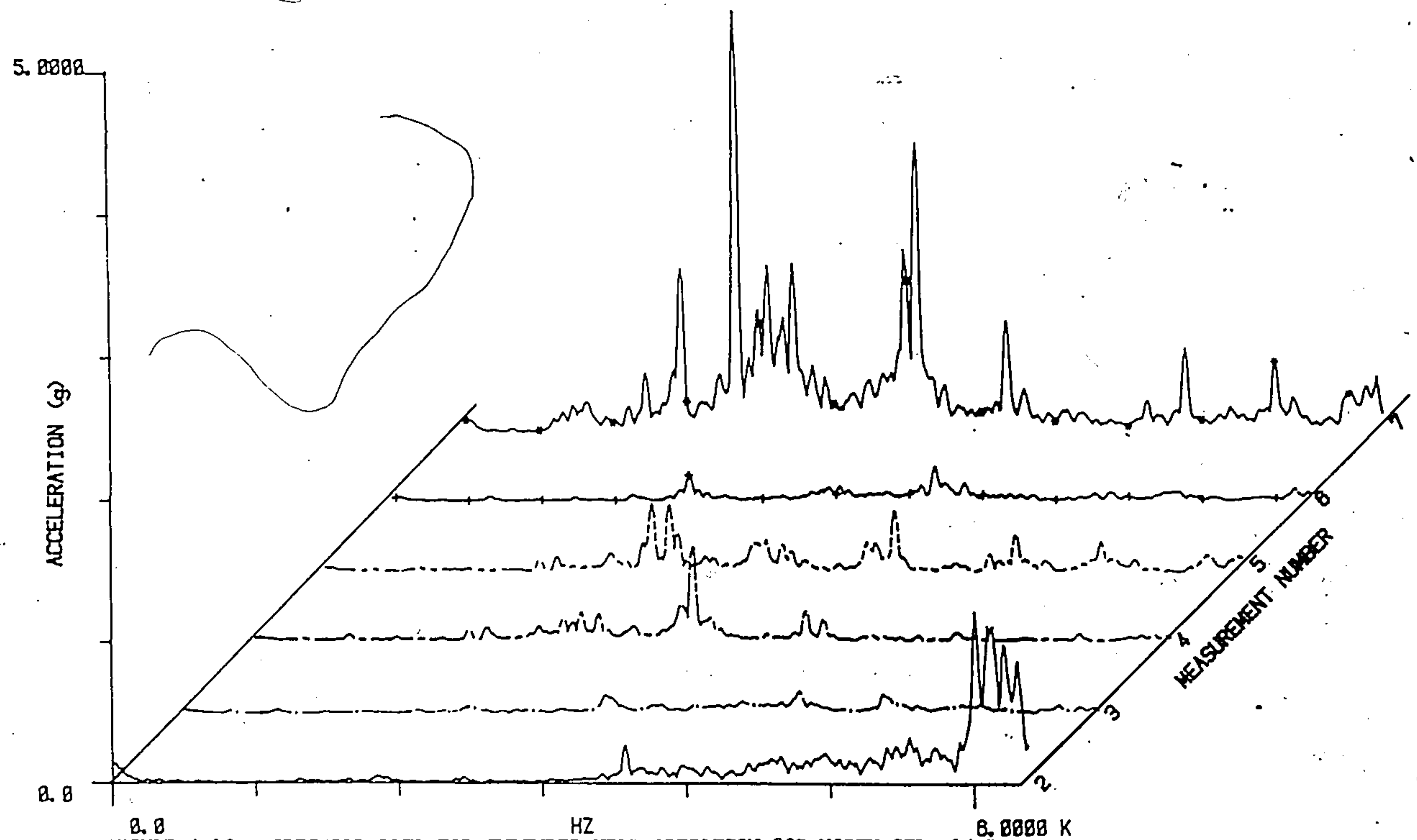


FIGURE 4.13: SPECTRAL DATA FOR CYLINDER HEAD OPERATION 20D NORTH STA. 14 R.
(MEASURING POSITION 1)

components. Measurement 2 indicates a spectrum associated with bearing failure. Note the high spectral components around 6000 Hz and the large amount of sideband energy associated with the spectrum. The bearing seized a short time after this measurement was taken. Measurement 3 shows the spectrum associated with the rebuilt spindles which replaced the failed unit. Measurement 7 was taken just prior to a second failure of this unit.

It is worth noting that for this particular machining station, the overall vibration level (see Figure 4.2b) is also quite a sensitive indicator of impending failure. This is not necessarily the case in all applications, however, it is obvious that for this unit, acceleration is a very good parameter for prediction of failure. Furthermore, the data in this figure have the classic "bathtub" shape associated with overall vibration levels as a mechanical system heads toward failure.

The rapid failure of this unit was due to the inability of the air purge system to keep the cutting fluid out of the front bearings of the spindles. This caused the "washing away" of the bearing lubricant and hence failure resulted from lubricant starvation.

The data associated with Operation 60 South, Station 17R, Camshaft (see Figure 4.5), provided some unique insights into vibration monitoring analysis. This machining station had at least one component failure as evidence by the low frequency "growl" associated with its operation. The decision was made to monitor this station in an attempt to determine the specific failure component. The initial reaction was that a bearing failure had occurred. This was an opinion based on the type of noise generated during operation and a review of the assembly design. The results of vibration measurements are illustrated in Figure 4.5 and the corresponding spectra for one measuring position in Figure 4.14. The results were not at all what was initially expected. The overall acceleration levels (see Figure 4.5) shows variation with time rather than steadily increasing levels as would normally be expected during bearing failure. By measurement number 6 (see Figure 4.14), there is predominant vibration component at about 375 Hz with little high frequency energy evident.

This puzzling data was clarified when the spindle was disassembled and it was found that three front bearings had been installed backwards. This resulted in a catastrophic failure of the bearings with two balls actually split in half. The low frequency vibration at about 375 Hz was obviously associated with the split ball oscillation frequency.

In this case, it was indicated that when the monitoring process had begun, the catastrophic failures had already

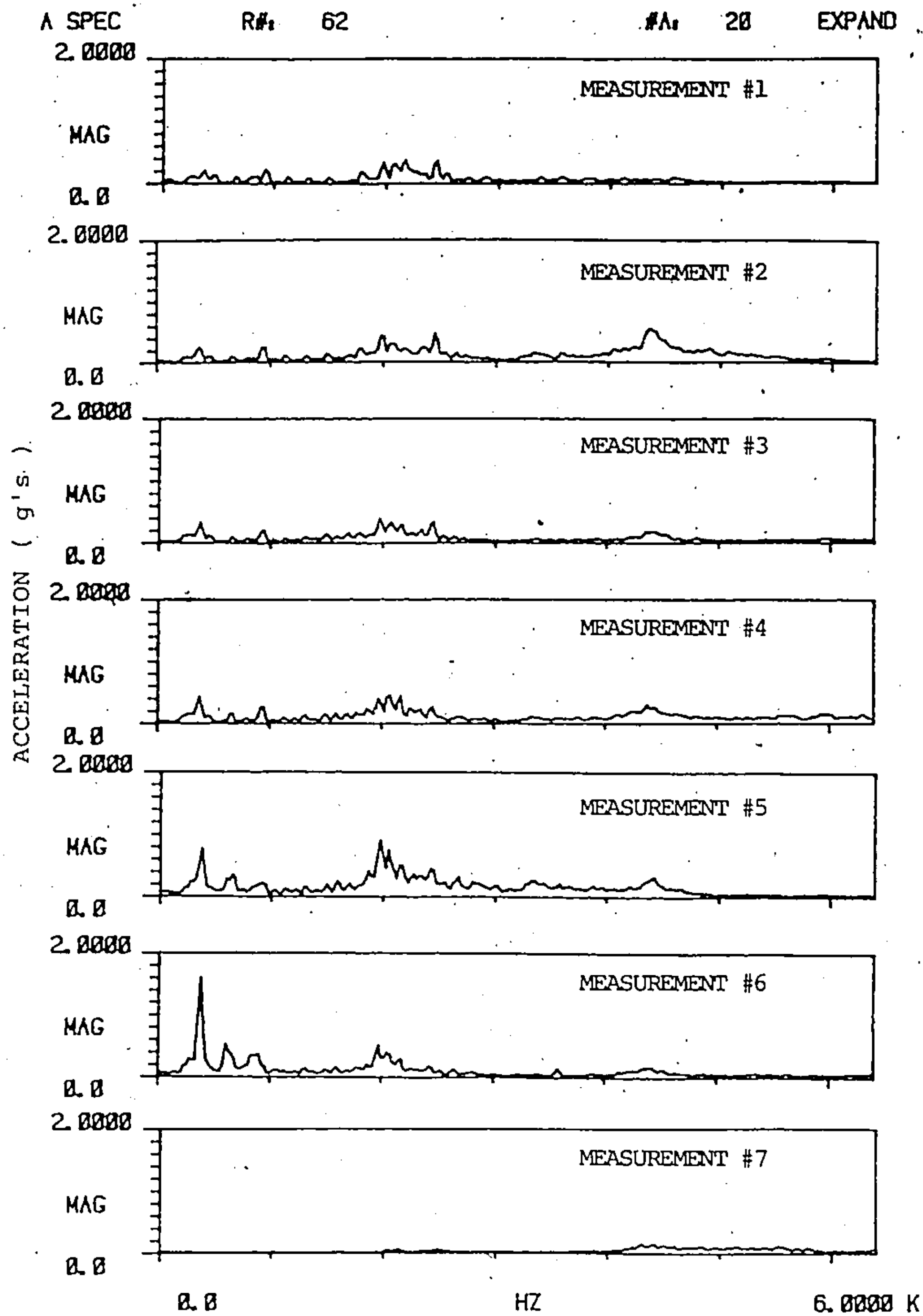


FIGURE 4.14: SPECTRAL DATA FOR CAMSHAFT OPERATION 60 SOUTH STA. 17R. (MEASURING POSITION 1)

occurred. The signs associated with developing bearing failure were not evident because failure was already complete. However, if monitoring had begun early enough, failure development would have been evident and remedial measures could have been taken.

It is obvious from this study that vibration monitoring techniques are indeed feasible for use on the high volume multistation transfer machines employed in a typical automotive engine plant. It has been shown that repeatable vibration measurements are possible and that damage of components can be detected prior to failure. The vibration monitoring technique has the potential for detecting even more intricate failure mechanisms.

In addition, representative overall vibration levels, as well as frequency spectra from designated operations, have been obtained for use as "baseline" spectra in any future vibration monitoring system that may be installed at the Ford Essex Engine Plant.

V. IDENTIFICATION OF BEARING DEFECTS IN A SINGLE SPINDLE MACHINING STATION

5.1 Introduction

As a result of successful "in plant" manual vibration monitoring, the study of a series of controlled bearing failure tests was performed in order to determine the most suitable vibration analysis techniques, both in the time domain and frequency domain, for identifying the types of bearing failure. The advantages and disadvantages of these analysis techniques are also reviewed.

All of the vibration data reported in this section was obtained from an experimental test rig which consisted of a typical single spindle machining station (see Figure 5.1). The defects induced on the bearing included those in the outer race, in the inner race, a ball and finally a combination of all three. The spindle was running at a speed of about 1680 rpm and was "idling" (not cutting metal) when measurements were taken.

COLOURED PICTURES
Images en couleur

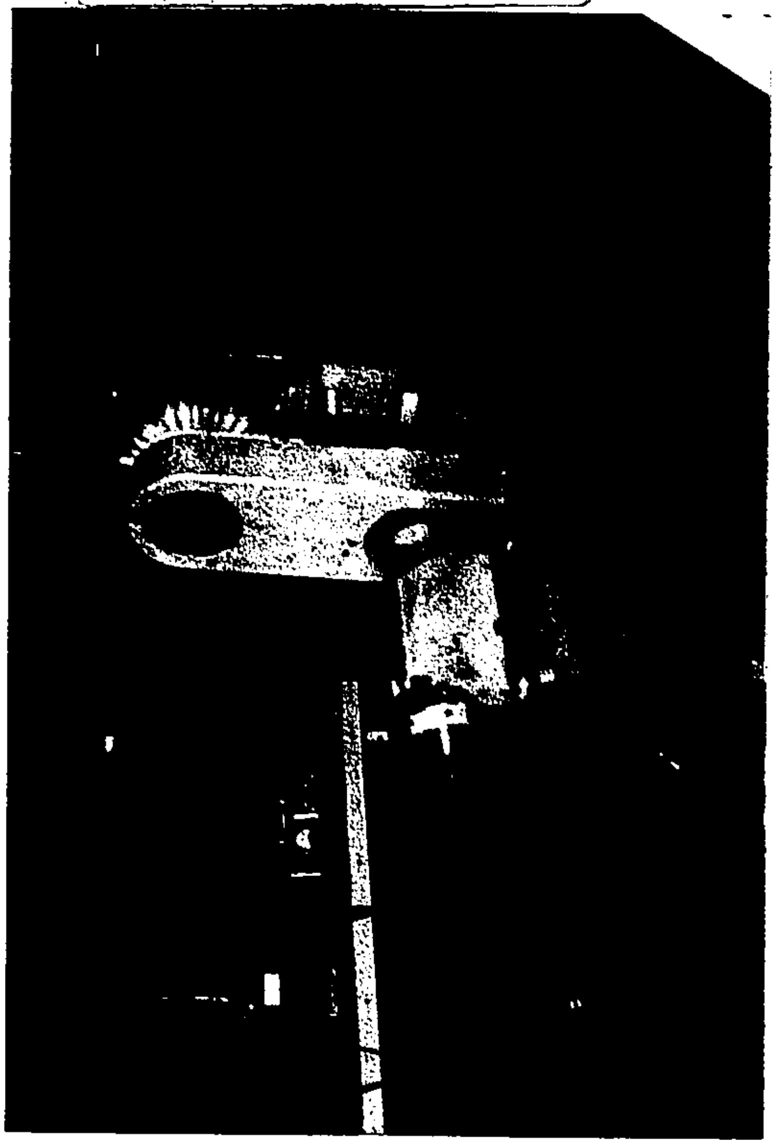


FIGURE 5.1: A SINGLE SPINDLE MACHINING STATION

5.2 Instrumentation

The following instrumentation was used in the tests:

a. Bruel and Kjaer 2209, Sound Level Meter,

b. Bruel and Kjaer 2635, Charge Amplifier,

c. Bruel and Kjaer 4333, Accelerometer,

d. Gould OS 4020, Digital Storage Oscilloscope,

e. Hewlett Packard 5423A, Structural Dynamics Analyzer,

f. Hewlett Packard 7045A, X-Y Plotter,

g. Hewlett Packard 9872B, Digital Plotter,

h. Nagra IV-SJ, Tape recorder,

i. Piezotronics, Inc., PCB Model 303A02, Accelerometer,

j. Piezotronics, Inc., PCB Model 408B06, Power Supply.

Brief descriptions of these intruments are found in Appendix C.

The schematic of the experimental setup is illustrated in Figure 5.4.

5.3 Measurement Methodology

The single spindle test rig (see Figure 5.1) was used to determine if known bearing defects could be reliably identified using vibration monitoring techniques. The spindle layout is shown in Figure 5.3. The assembly of the spindle is illustrated in Figures 5.2a and 5.2b.

COLOURED PICTURES
Images en couleur

FIGURE 5.2a: SPINDLE CASING.

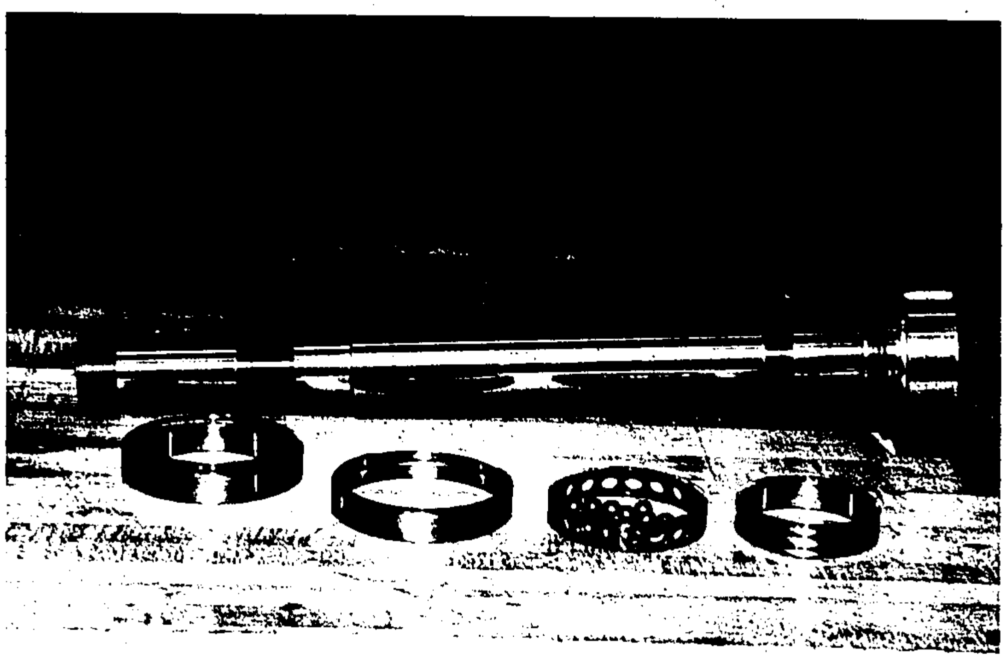
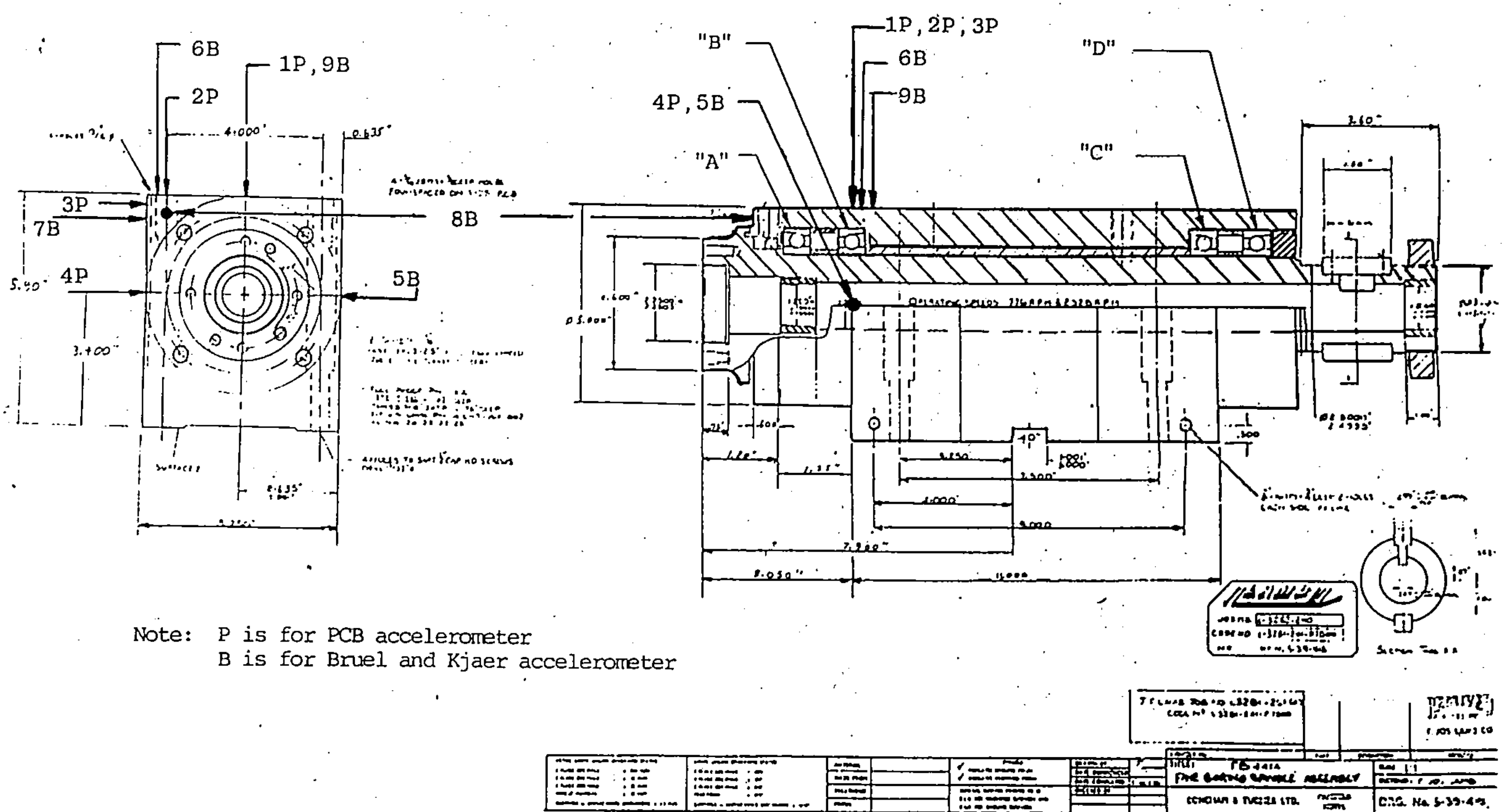


FIGURE 5.2b: THE SHAFT AND BEARING ASSEMBLY



Note: P is for PCB accelerometer
B is for Bruel and Kjaer accelerometer

FIGURE 5.3: SPINDLE LAYOUT INCLUDING ALL THE MEASURING POSITIONS

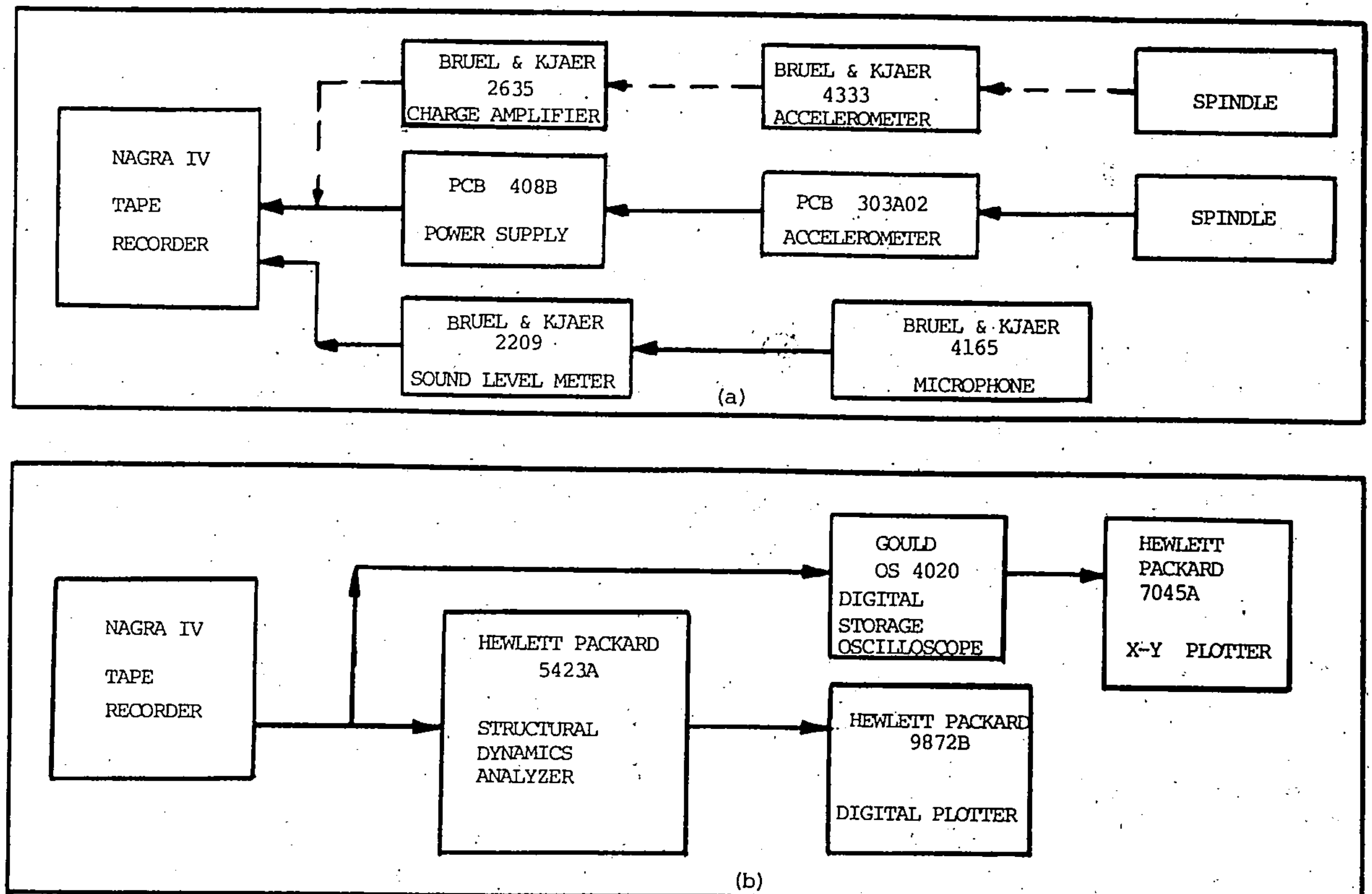


FIGURE 5.4: THE SCHEMATIC OF EXPERIMENTAL SET UP (a) IN-PLANT MEASUREMENTS (b) IN LABORATORY ANALYSIS

The schematic of the experimental set up is shown in Figure 5.4. The spindle was operating at an idle speed of 1680 rpm. There was no cutting process involved, thus no significant radial or thrust loads were present.

As illustrated in Figure 5.3, there were four thrust bearings labelled "A" to "D" in the spindle. It was decided that the bearing "B" was to be used for testing because of ease of removal and assembly. The test bearing "B" was an RHP precision bearing with code number 7013 TU EP3.

Two different accelerometers were used to measure vibration (acceleration) at various measuring positions as indicated in Figure 5.3. This was done to take advantage of the two different frequency response characteristics of the accelerometers thus permitting a particular range of interest to be studied using the most appropriate transducer. Also, to a limited extent, the two accelerometers could be used as back up for one another.

The accelerometers were directly stud mounted so as to minimize the mounting resonance effects over the desired frequency range of interest. Other mounting methods, such as magnetic base mounting and wax mounting, were found to reduce the frequency response of the measuring system.

The vibration (acceleration) signal was conditioned and then recorded on a scientific quality reel to reel tape recorder. The frequency response of the measuring system, when using the PCB accelerometer, extended from 2 Hz to 20 kHz (± 10%). The frequency response of the system with the Bruel and Kjaer accelerometer extended from 2 Hz to 21 kHz (± 10%). Furthermore, the resonant frequency of PCB accelerometer is 70 kHz while that of the Bruel and Kjaer accelerometer is only 36 kHz. A sound level meter was used to cue in the measuring instruments set up as well as for neccessary comments.

The recorded acceleration signal was later analyzed in the laboratory using both an oscilloscope and an FFT analyzer to obtain time and frequency domain data respectively. The frequency analysis was carried out in two ranges of interest. The low range was 0 to 1.6 kHz while the high range extended from 0 to 25.6 kHz.

The analog to digital converter range settings of the FFT analyzer were always set at optimum values when analyzing the recorded vibration signal. This was done to minimize the error due to leakage. A flat-top window was also applied to the signal to further reduce leakage error to a minimum and also to obtain better amplitude resolution. All the frequency spectra were averaged to reduce the effects of noise and random components. Only fifty averages were used since a larger number was found not to provide a significant enhancement of the spectrum. The frequency spectra were stored on digital tape and could be recalled at any time for analysis.

The digital plotter was used to obtain hard copies of the results.

An analog X-Y plotter was used to obtain hard copies of the time domain data (amplitude versus time) from the oscilloscope.

During the course of this study, tests were carried out to investigate the followings:

A. the effects of removing and assembling the same test bearing,

B. the effects of induced outer race defect (see Figure 5.5),

C. the effects of induced ball defect (see Figure 5.6),

D. the effects of induced inner race defect (see Figure 5.7) and

E. the effects of multiple induced defects (see Figure 5.8).

For test A, the time domain data and frequency domain data were obtained prior to the removal of the test bearing "B" and after the reassembly of the same bearing.

For test B to E, three sets of measurements were obtained when a new test bearing was in place. Then the test bearing was removed and the specified defect was induced before being reassembled. Three sets of measurements were then taken with the induced defect present. Table 5.1 summarizes all measurements obtained for the above tests.

COLOURED PICTURES
Images en couleur

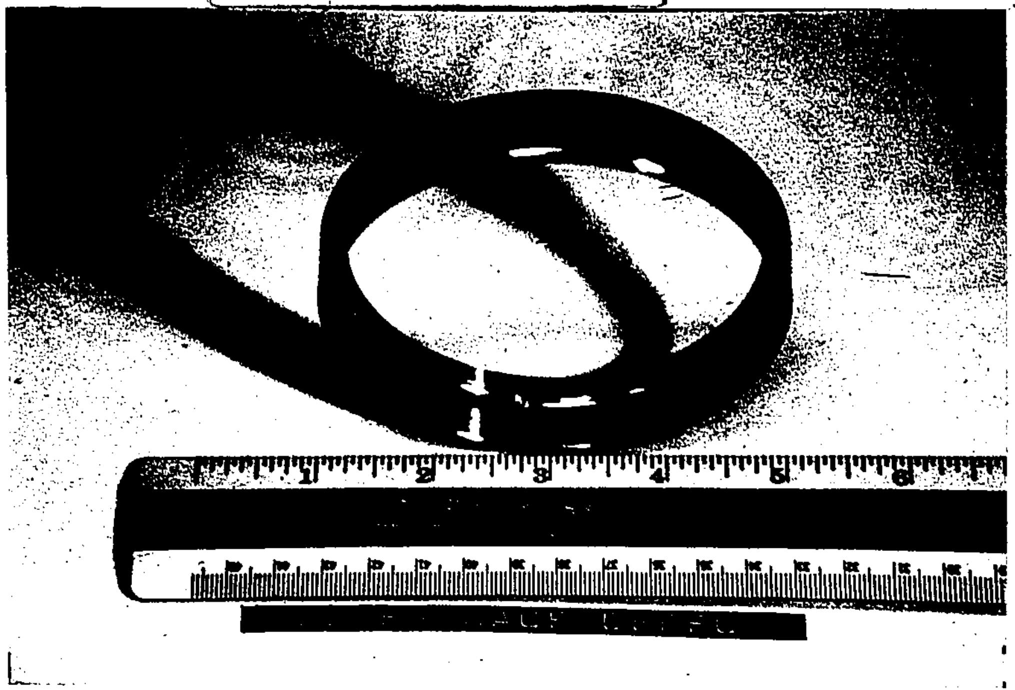


FIGURE 5.5: AN INDUCED OUTER RACE DEFECT

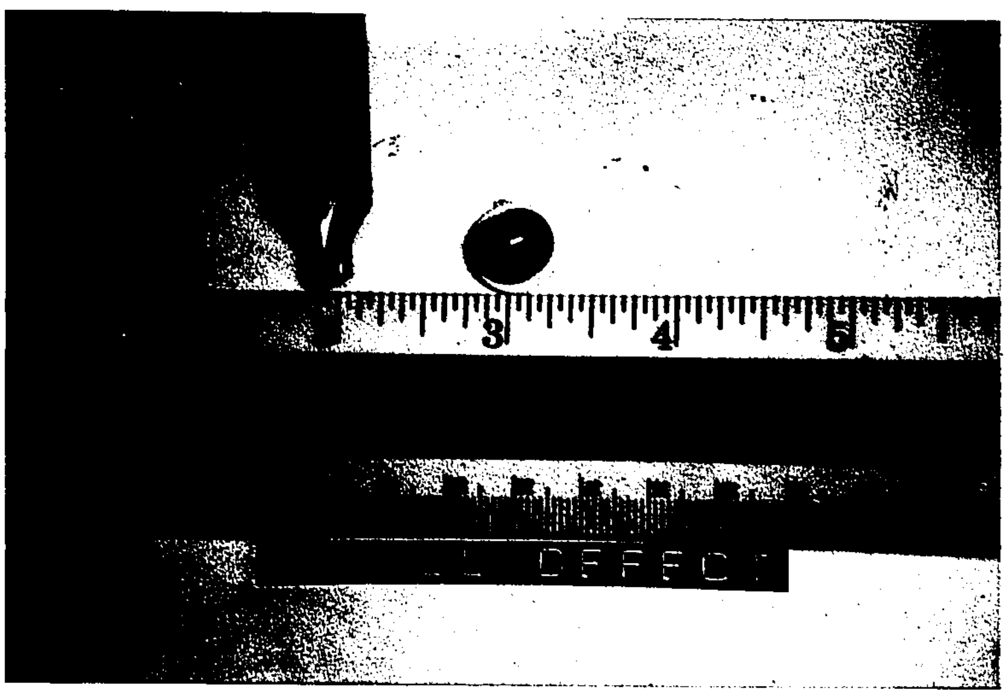


FIGURE 5.6: AN INDUCED BALL DEFECT

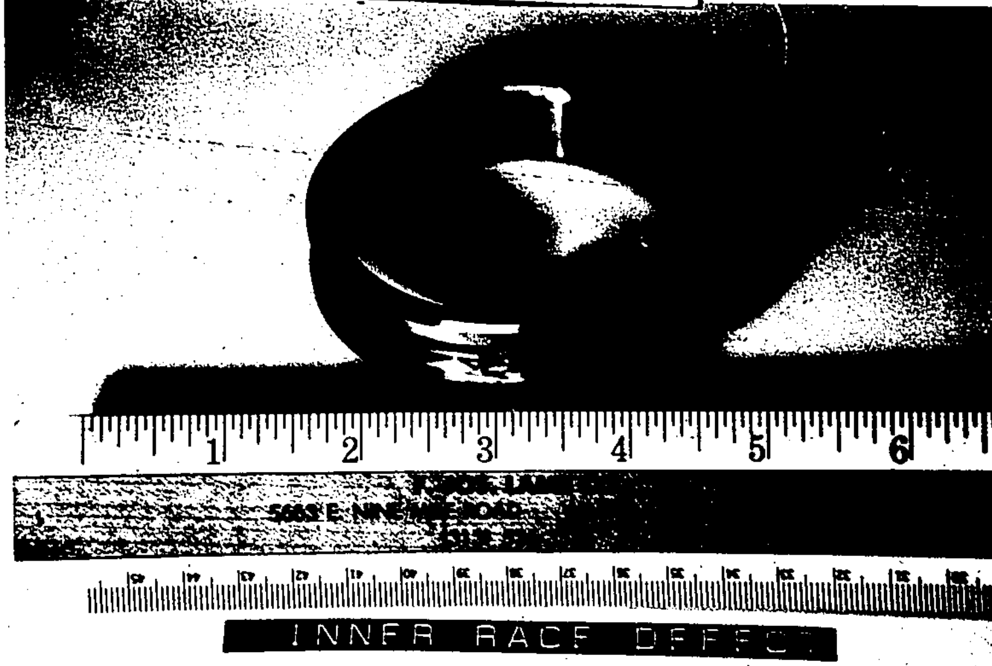


FIGURE 5.7: AN INDUCED INNER RACE DEFECT

FIGURE 5.8: INDUCED MULTIPLE DEFECTS

DATE	MEASUREMENT TYPE	SPINDLE IN OPERATION (HOURS)
09/15/1983	BASELINE	10
09/15/1983	REMOVAL & INCORRECTLY ASSEMBLED	11
09/16/1983	REMOVAL & CORRECTLY ASSEMBLED	12
11/02/1983	BASELINE	12
11/02/1983	BASELINE	22
11/03/1983	BASELINE	40
11/09/1983	INNER RACE DEFECT	24
11/10/1983	INNER RACE DEFECT	48
11/10/1983	INNER RACE DEFECT	52
11/14/1983	BASELINE	12
11/15/1983	BASELINE	25
11/16/1983	BASELINE	46
11/21/1983	BALL DEFECT	10
11/22/1983	BALL DEFECT	30
11/22/1983	BALL DEFECT	40
10/17/1983	BASELINE	10
10/20/1983	BASELINE	34
10/24/1983	BASELINE	50
11/23/1983	OUTER RACE DEFECT	12
11/23/1983	OUTER RACE DEFECT	18
11/24/1983	OUTER RACE DEFECT	30
11/25/1983	MULTIPLE DEFECTS	18
11/26/1983	MULTIPLE DEFECTS	36
11/28/1983	MULTIPLE DEFECTS	84

TABLE 5.1: SUMMARY OF ALL THE VIBRATION MEASUREMENTS

It includes the date, measurement types and the number of hours the spindle was in operation.

5.4 Calculations

Generally, there are five main frequencies associated with defective bearings. Various data obtained from the bearing manufacturer's design specifications and operational variables associated with the machine itself, are used in relatively simple formulas to calculate the numerical values of these frequencies.

A machine (e.g. a spindle) with a defective bearing can generate vibration at five main frequencies. They are:

a. the rotational unit frequency, f_R, which is generated by the speed of the rotating unit. It is generally caused by residual unbalance and / or eccentricity in the rotating unit.

b. the fundamental train frequency, f_A, which is the rotating speed of balls and cage assembly. It is seldom encountered except when some defects affect the rotation of the train.

c. the ball spin frequency, f_B, which is generated when a defect on a ball strikes the raceways.

d. the ball pass frequency of the outer race, f_O, which is generated as the balls pass over a defect on the outer race.

e. the ball pass frequency of the inner race, f_I, which is generated as the balls pass over a defect on an inner race.

The following definitions and data are required to compute the five main frequencies generated by defective bearings (see section 3.2):

f_R = rotational shaft frequency = $\frac{rpm}{60} = \frac{1680}{60}$ = 28 Hz

N_B = number of balls = 19

D_P = pitch diameter = 82.676 mm

ϕ = contact angle = 20^0

d_B = ball diameter = 10.318 mm

From section 3.2:

Equation (3.2.10) $$f_B = \frac{f_R}{2}\left[1 - \left(\frac{d_B}{D_P}\cos\phi\right)^2\right]$$

$$f_B = \frac{28}{2}\left[1 - \left(\frac{10.318}{82.676}\cos 20^\circ\right)^2\right]$$

$$\underline{f_B = 110.64 \text{ Hz}}$$

Equation (3.2.11) $$f_A = \frac{f_R}{2}\left[1 - \left(\frac{d_B}{D_P}\cos\phi\right)\right]$$

$$= \frac{28}{2}\left[1 - \left(\frac{10.318}{82.676}\cos 20^\circ\right)\right]$$

$$\underline{f_A = 12.36 \text{ Hz}}$$

Equation (3.2.12)

$$f_O = \frac{N_B f_R}{2}\left[1 - \left(\frac{d_B}{D_P}\cos\phi\right)\right]$$

$$= \frac{19(28)}{2}\left[1 - \left(\frac{10.318}{82.676}\cos 20^\circ\right)\right]$$

$$\underline{f_O = 234.81\ \text{Hz}}$$

Equation (3.2.13)

$$f_I = \frac{N_B f_R}{2}\left[1 + \left(\frac{d_B}{D_P}\cos\phi\right)\right]$$

$$= \frac{19(28)}{2}\left[1 + \left(\frac{10.318}{82.676}\cos 20^\circ\right)\right]$$

$$\underline{f_I = 297.19\ \text{Hz}}$$

5.5 Results and Discussions

The results of the tests conducted are discussed individually using both the time domain data and the frequency domain data

5.5.1 The Effects of Removing and Reassembling the Same Test Bearing

Figures 5.9a and 5.9b illustrate frequency spectra obtained prior to the removal of the test bearing and that obtained after the same bearing was simply removed and then replaced in the spindle. It is obvious from these Figures that the removal and reassembly of the test bearing had no significant effect on the frequency spectra.

Figures 5.10a and 5.10b show the spectra of a new test bearing correctly installed versus a new test bearing installed with slightly higher preload than recommended by the bearing manufacturer. These spectra clearly indicate that incorrect installation could significantly increase the vibration levels.

Hence, it is apparent that if a test bearing has a defect introduced after taking baseline measurements and if it is then reassembled correctly, any changes in the frequency spectrum are attributed to the induced defect.

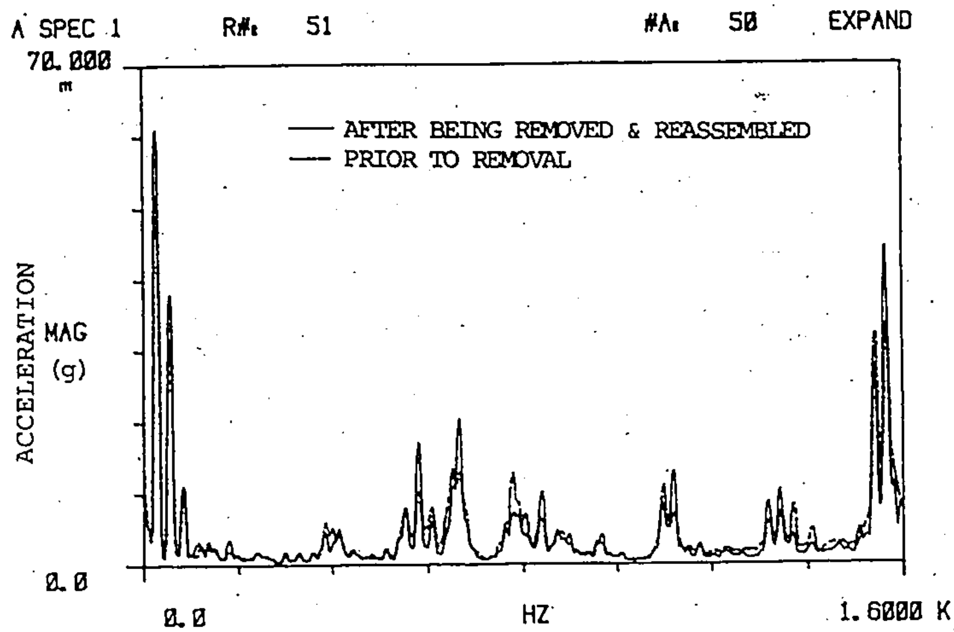


FIGURE 5.9a: COMPARISON OF FREQUENCY SPECTRA OF BEARING PRIOR TO REMOVAL AND AFTER BEING REMOVED AND REASSEMBLED

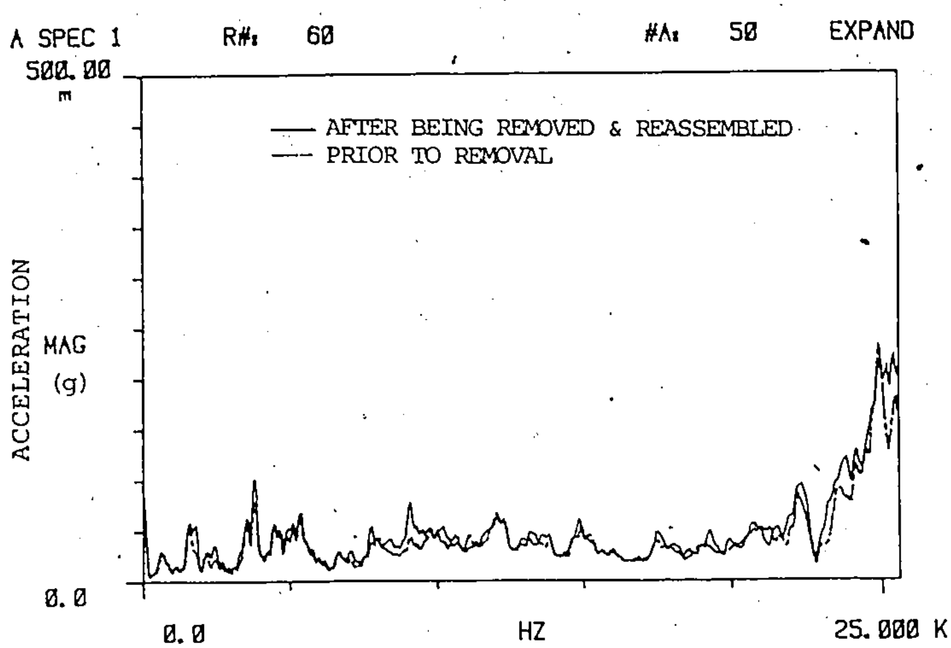


FIGURE 5.9b: COMPARISON OF FREQUENCY SPECTRA OF BEARING PRIOR TO REMOVAL AND AFTER BEING REMOVED AND REASSEMBLED

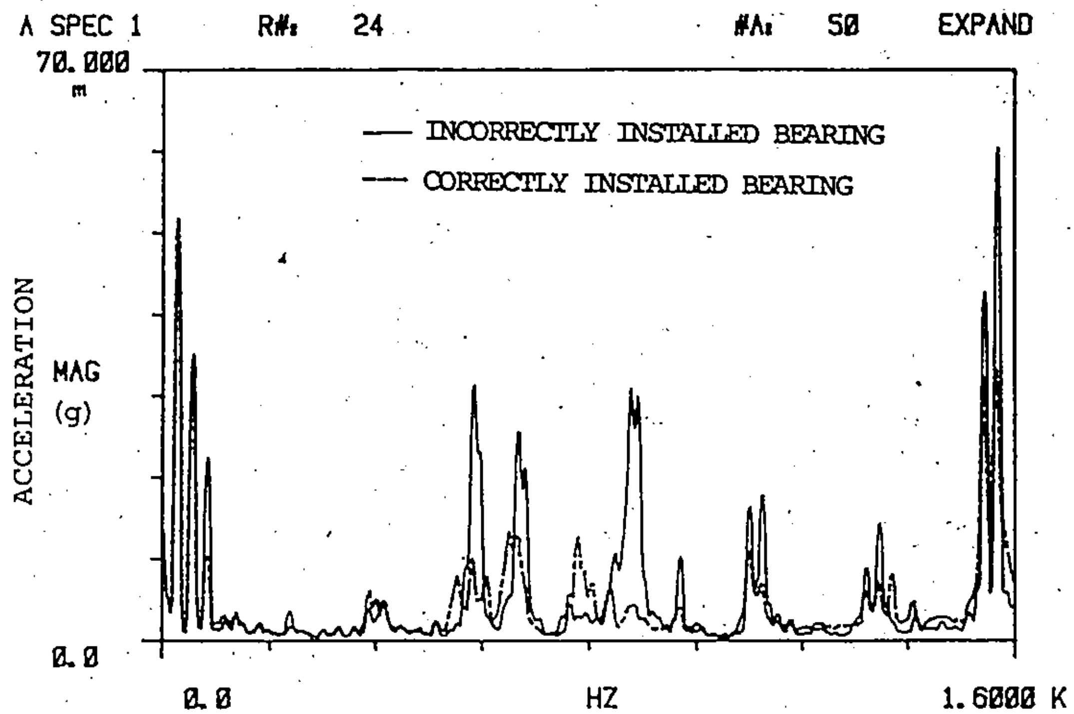


FIGURE 5.10a: COMPARISON OF FREQUENCY SPECTRA OF INCORRECTLY INSTALLED BEARING AND CORRECTLY INSTALLED BEARING

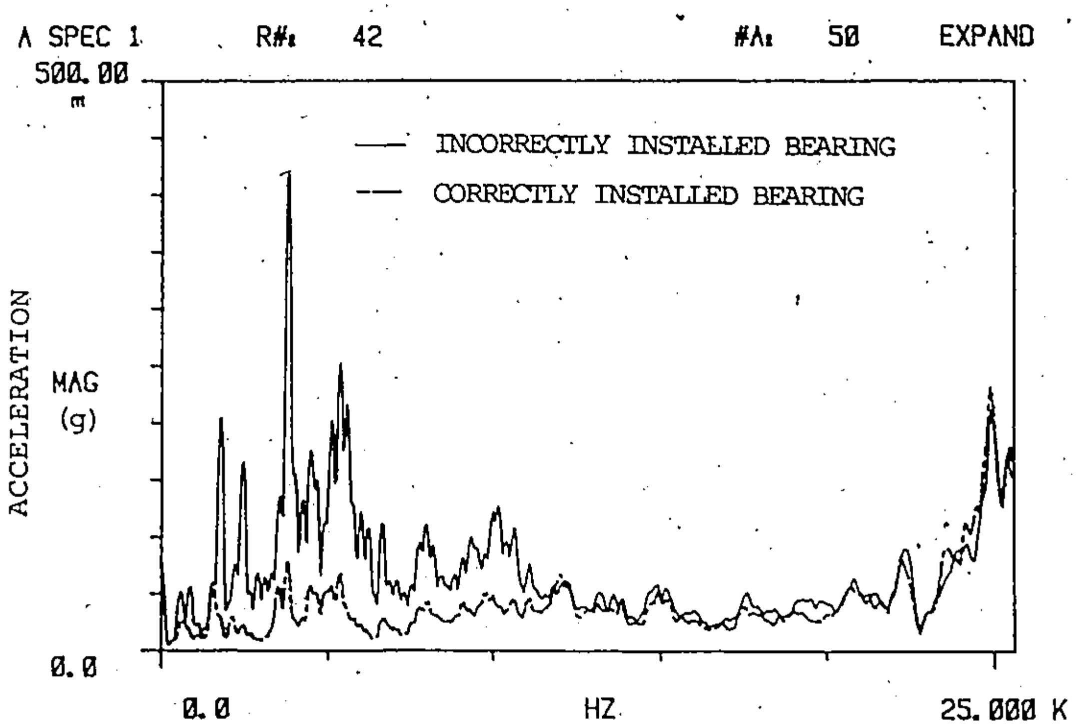


FIGURE 5.10b: COMPARISON OF FREQUENCY SPECTRA OF INCORRECTLY INSTALLED BEARING AND CORRECTLY INSTALLED BEARING

5.5.2 The Effects of Induced Outer Race Defect

The ball pass frequency of the outer race (BPFO) is generated by the spindle with a bearing that has an induced outer race defect. This frequency is produced as the balls pass over a defect on the outer race. This phenomenon is illustrated in Figures 5.11b to 5.11d. They illustrate typical results for an induced outer race defect. Furthermore, results of the remaining measuring positions are found in Appendix D.

Based on the parameters of the experimental set up, the ball pass frequency of the outer race was computed to be 234.81 Hz and was found experimentally (from the frequency spectra) to have approximately the same value. The BPFO generated was roughly equal to 40 percent of the product of the number of balls and the revolution per second of the shaft (212.8 Hz). This is true because 40 percent of the balls pass over the defect on the outer race during each spindle revolution.

Besides the fundamental ball pass frequency of the outer race, harmonics of the BPFO were also generated by an induced outer race defect.

Generally, the low frequency analysis (0 - 1.6 kHz) indicates that the BPFO and its harmonics were generated wherever the vibration measurements were taken. However,

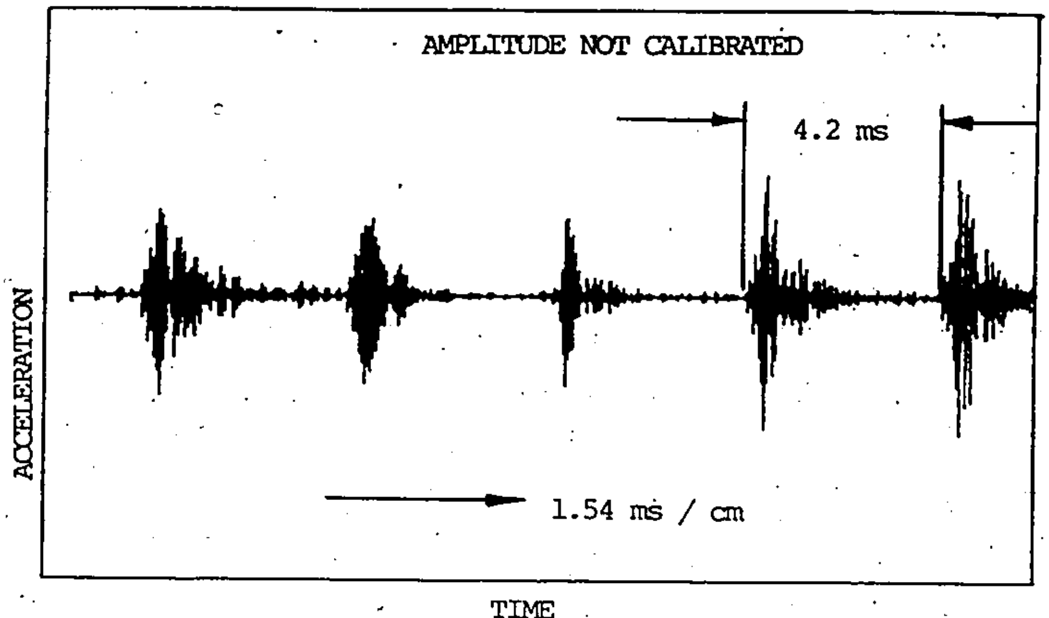


FIGURE 5.11a: ACCELERATION VERSUS TIME FOR OUTER RACE DEFECT (POSITION 5B)

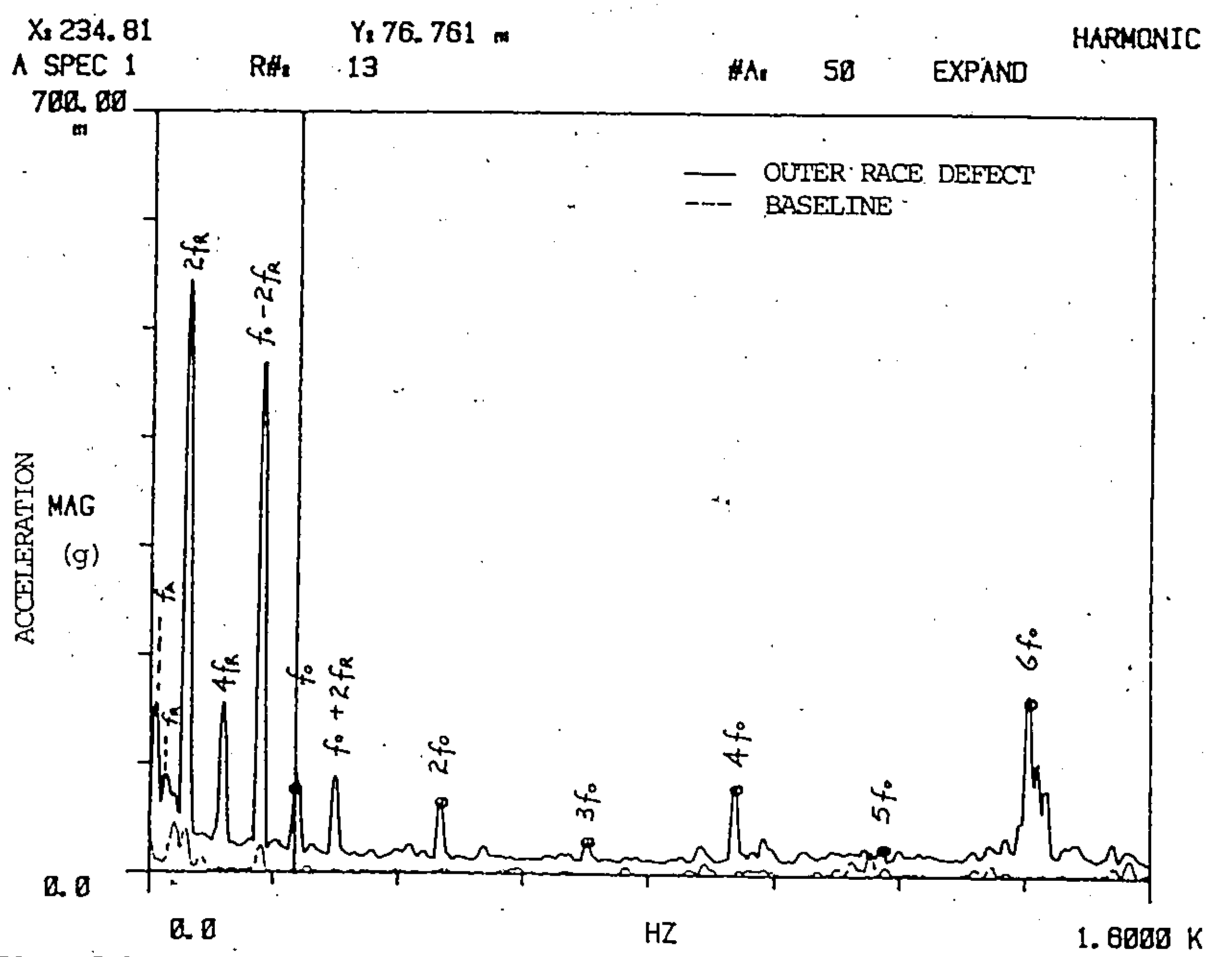


FIGURE 5.11b: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B, 0 – 1.6kHz

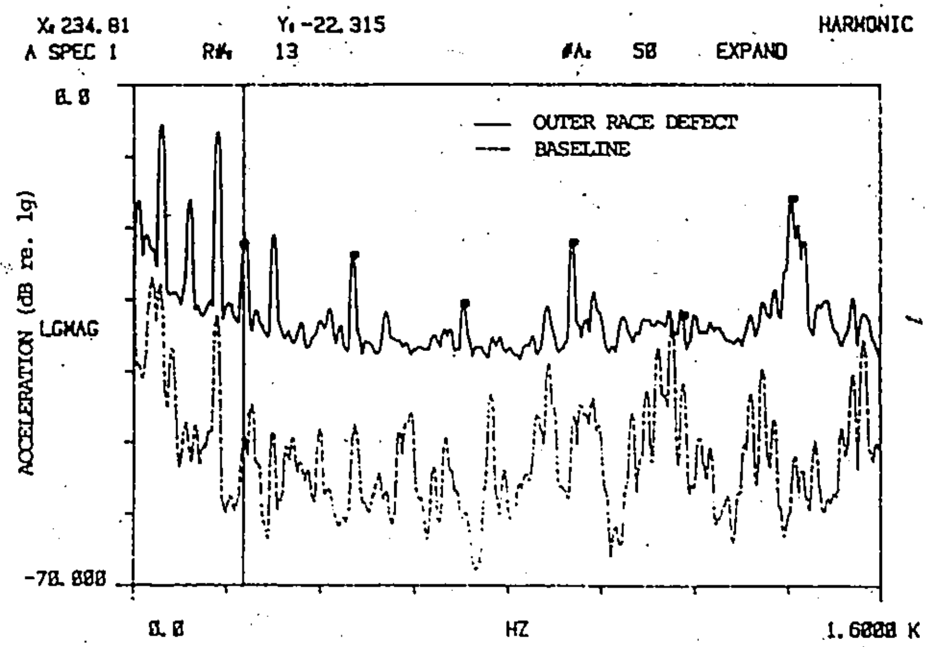


FIGURE 5.11c: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B (DECIBEL SCALE), 0 - 1.6kHz

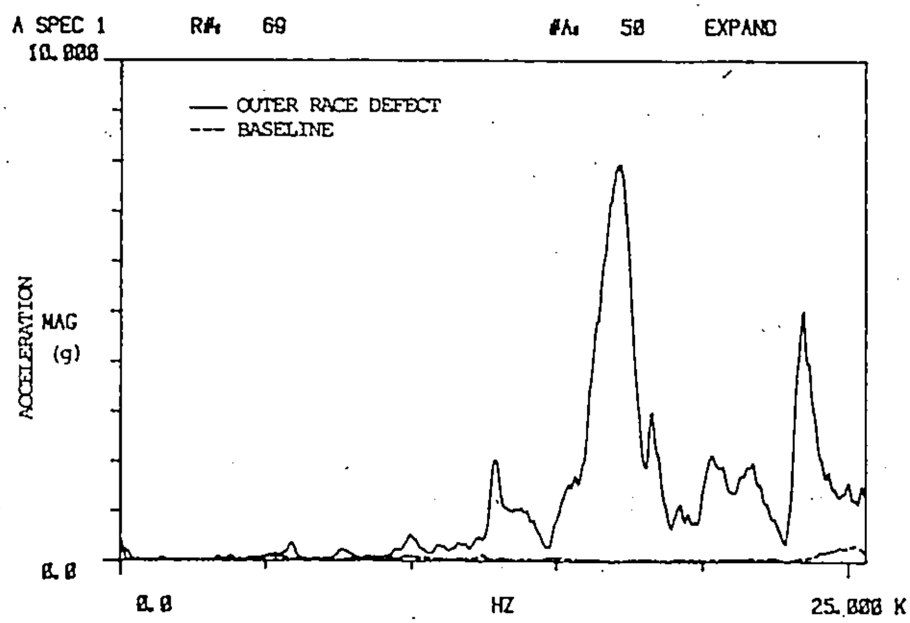


FIGURE 5.11d: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B, 0 - 25.6kHz

the differences in amplitudes between the induced outer race defect spectra and baseline spectra vary considerably among the measuring positions. Figure 5.11b, the linear spectrum and Figure 5.11c, the spectrum in decibel scale, illustrate all the frequencies generated by the outer race defect, which include the rotational shaft frequency, the fundamental train frequency, the ball pass frequency of the outer race and sums and differences of these frequencies.

The fundamental rotational frequency and its harmonics were also generated. Comparison of the outer race defect spectra and baseline spectra, Figure 5.11b indicates that there were large differences in amplitudes of the second harmonic of the fundamental rotational shaft frequency. This is mainly due to looseness because the induced defect on the outer race was large enough to allow movement of the rotating unit.

Furthermore, this also causes the BPFO to be modulated with the rotational shaft frequency. This modulation generates a side lobe to the ball pass frequency. The difference between the ball pass frequency of the outer race and the side lobe is equal to the multiple of the rotational shaft frequency as indicated by $f_O + 2f_R$ and $f_O - 2f_R$ in Figure 5.11b.

High frequency analysis (0 - 25.6 kHz) clearly shows the differences between the outer race defect spectra and the baseline spectra as illustrated in Figure 5.11d. Vast differences in the amplitudes of the spectra are found in the frequencies above 8.0 kHz. These results suggest that high frequency analysis is a very reliable means of detecting a defect on the outer race. The drawback to this technique is that the large bandwidth does not allow accurate determination of each of the frequency components due to the lack of frequency resolution.

Time domain analysis (amplitude versus time) indicates that there are clearly defined "pulses" generated as the balls pass over the induced defective outer race (see Figure 5.11a). The period of the pulses was found to be 4.2 milliseconds and hence the ball pass frequency of the outer race is calculated to be 238 Hz. The difference between this frequency and the BPFO computed in section 5.4 is approximately 1.4 percent, which is well within the range of measurement error. The amplitude versus time traces of each of the measuring positions indicate the presence of well defined pulses with constant frequencies. Note that the ordinate is not calibrated since comparisons of magnitudes are of primary interest.

Thus it is obvious that time domain analysis is also reliable in identifying an induced outer race defect.

5.5.3 The Effects of Induced Ball Defect

Theoretically, ball spin frequency is generated when a defect on the ball strikes the raceway. The frequency can be twice the ball pass frequency because the defect strikes both races. The frequency generated is seldom as high as these frequencies because the ball is not always in the load zone when the defect strikes. Energy is also lost when the defective ball strikes the inner race and dissipates energy through the shaft.

The spectra of an induced ball defect versus that of a new bearing for measuring position 6B do not show any significant difference in both the low frequency (see Figures 5.12b and 5.12c) and high frequency ranges (see Figure 5.12d). The spectra of all the remaining measuring positions are contained in Appendix E.

Time domain data (amplitude versus time) for the bearing with an induced ball defect is illustrated in Figure 5.12a. These results are very similar to those of a new bearing (baseline).

The reasons for the difficulty in detecting an induced ball defect were discussed previously. Furthermore, since averaging is used in the processing of the signal, the ball spin frequency and its harmonics (which seldom have high amplitude) tend to get buried by other frequency components of the "instantaneous spectra".

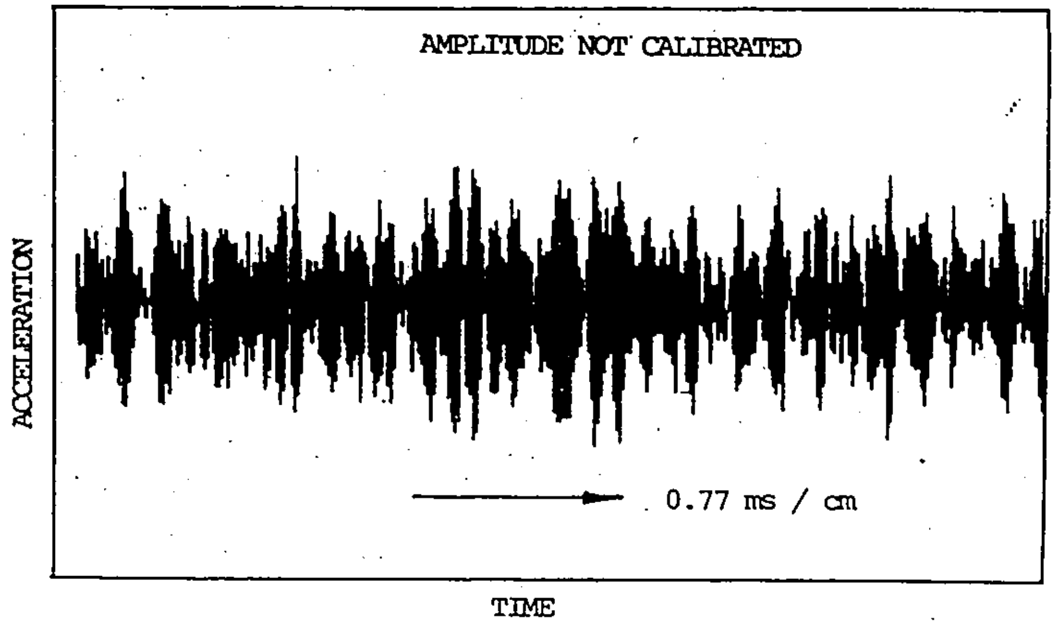


FIGURE 5.12a: ACCELERATION VERSUS TIME FOR BALL DEFECT (POSITION 6B)

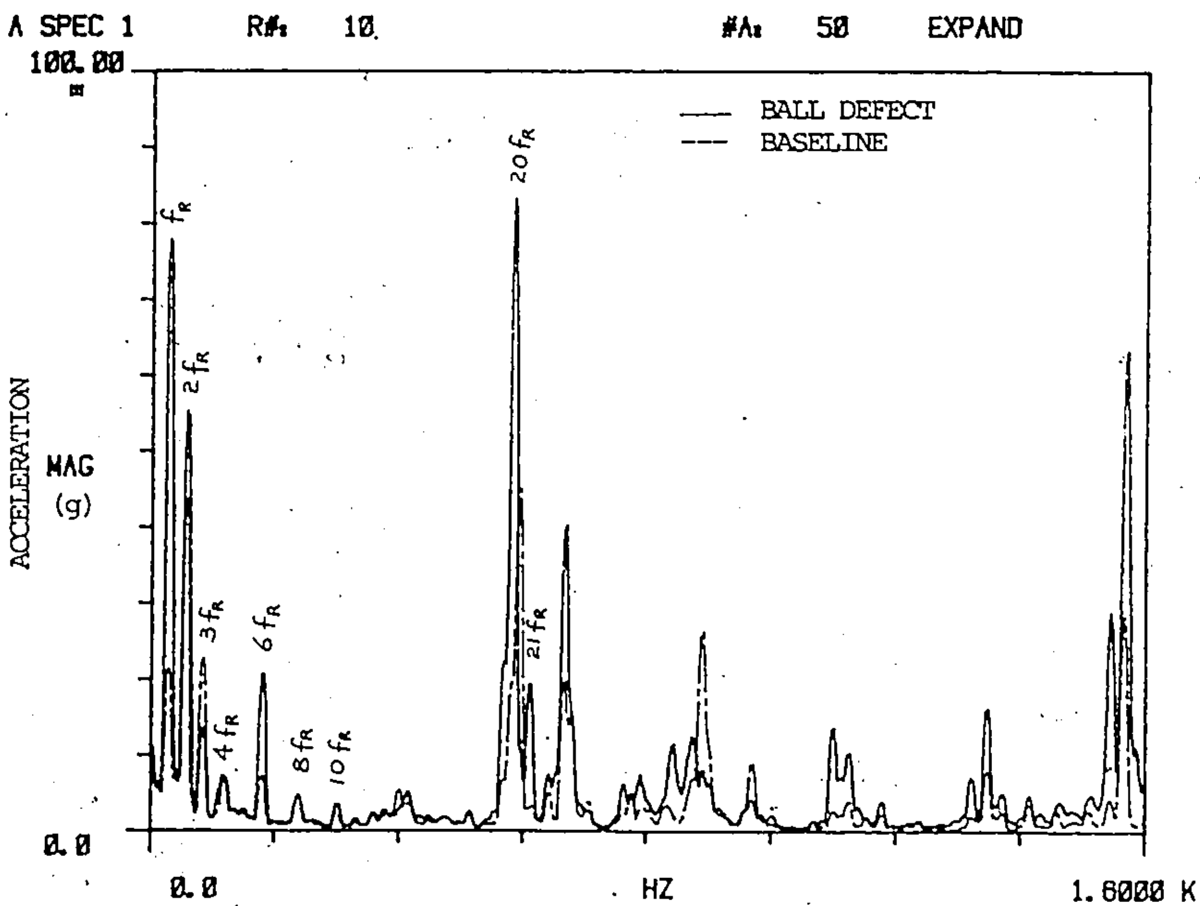


FIGURE 5.12b: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B, 0 - 1.6 kHz

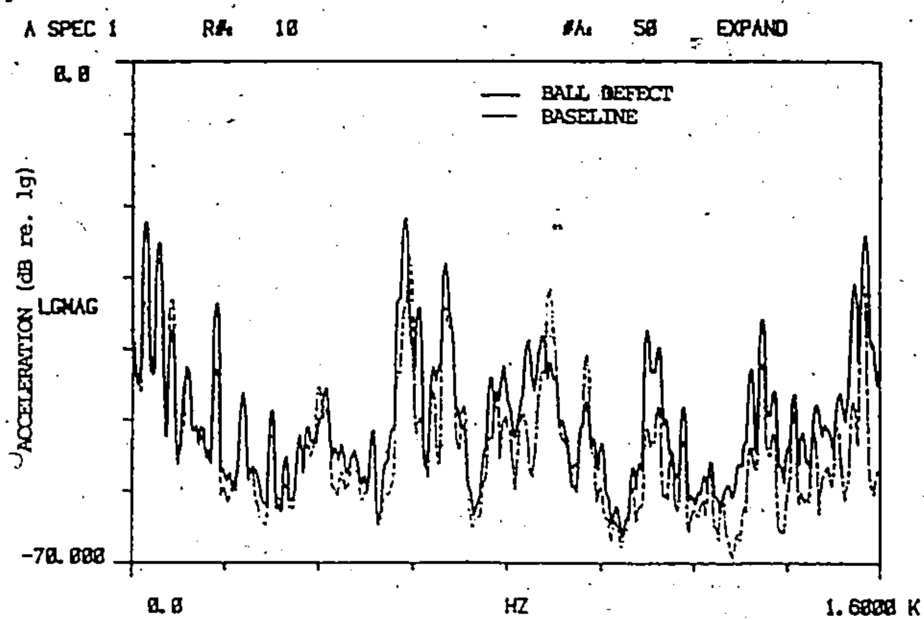


FIGURE 5.12c: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B (DECIBEL SCALE), 0 - 1.6kHz

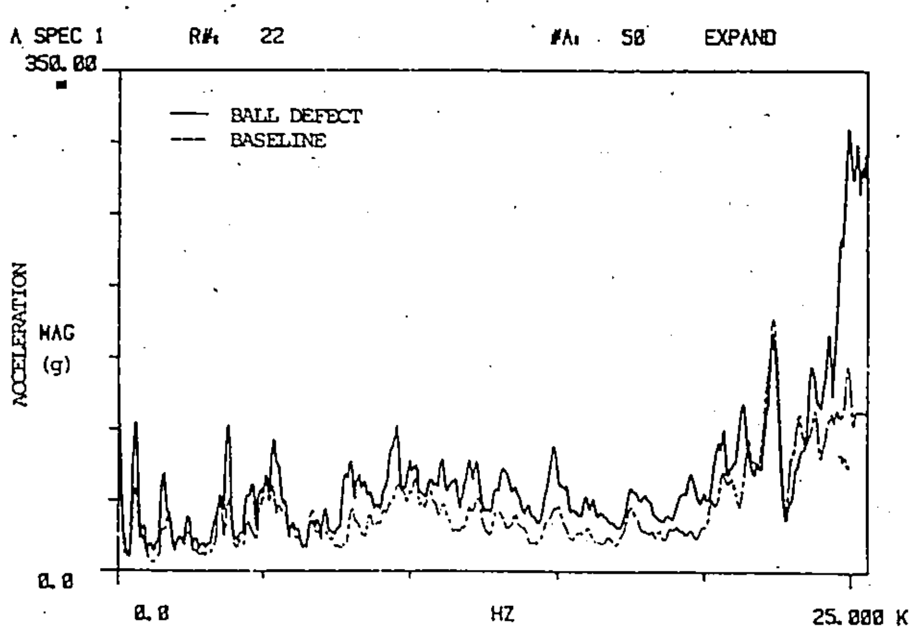


FIGURE 5.12d: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B, 0 - 25.6kHz

5.5.4 The Effects of Induced Inner Race Defect

The ball pass frequency of the inner race (BPFI) is generated as the balls pass over a defect on the inner race, as illustrated in Figures 5.13b and 5.13c, for measuring position 6B. The spectra of the remaining positions are listed in Appendix F. The BPFI was calculated to be 297 Hz and was found experimentally to be practically the same (see Figure 5.13b). This frequency is approximately equal to 60 percent of the product of the number of balls and the revolution per second of the shaft (319 Hz). This is true because 60 percent of the balls pass over the defect during a spindle revolution.

Besides the fundamental ball pass frequency of the inner race, its harmonics (especially the second and third harmonics) were also generated.

From the previously mentioned Figures, we notice that the even harmonics of the fundamental frequency are also generated. This is indicative of internal looseness of the bearing as the induced inner race defect caused the assembly tolerances to exceed those specified by the bearing manufacturer.

Figure 5.13b illustrates all the frequencies generated by an induced inner race defect. These frequencies include the rotational shaft frequency, the ball pass frequency of the inner race and sums and differences of these frequencies.

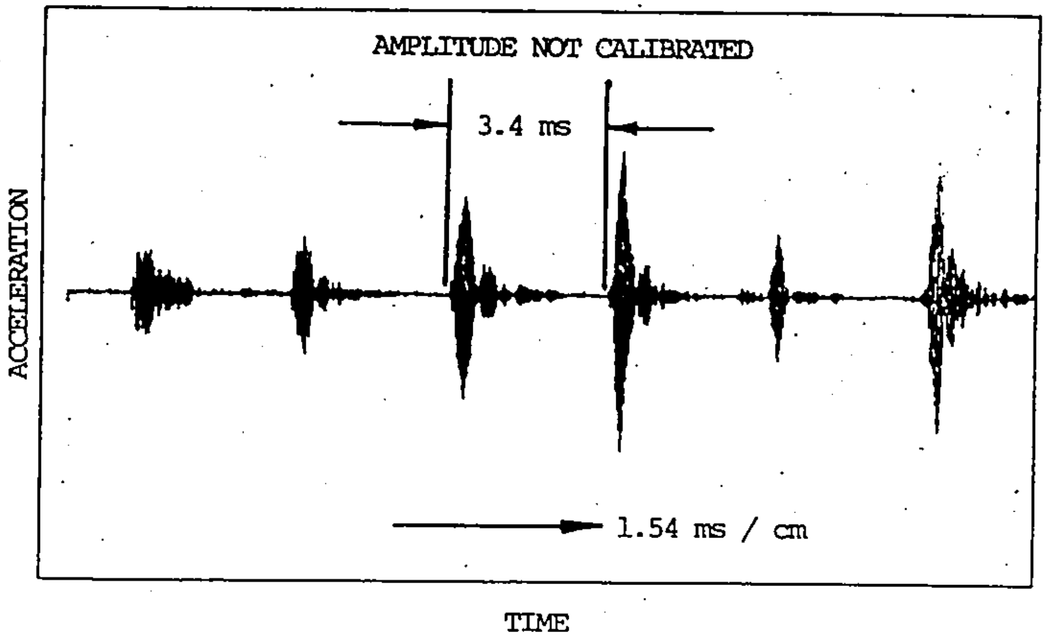


FIGURE 5.13a: ACCELERATION VERSUS TIME FOR INNER RACE DEFECT (POSITION 6B)

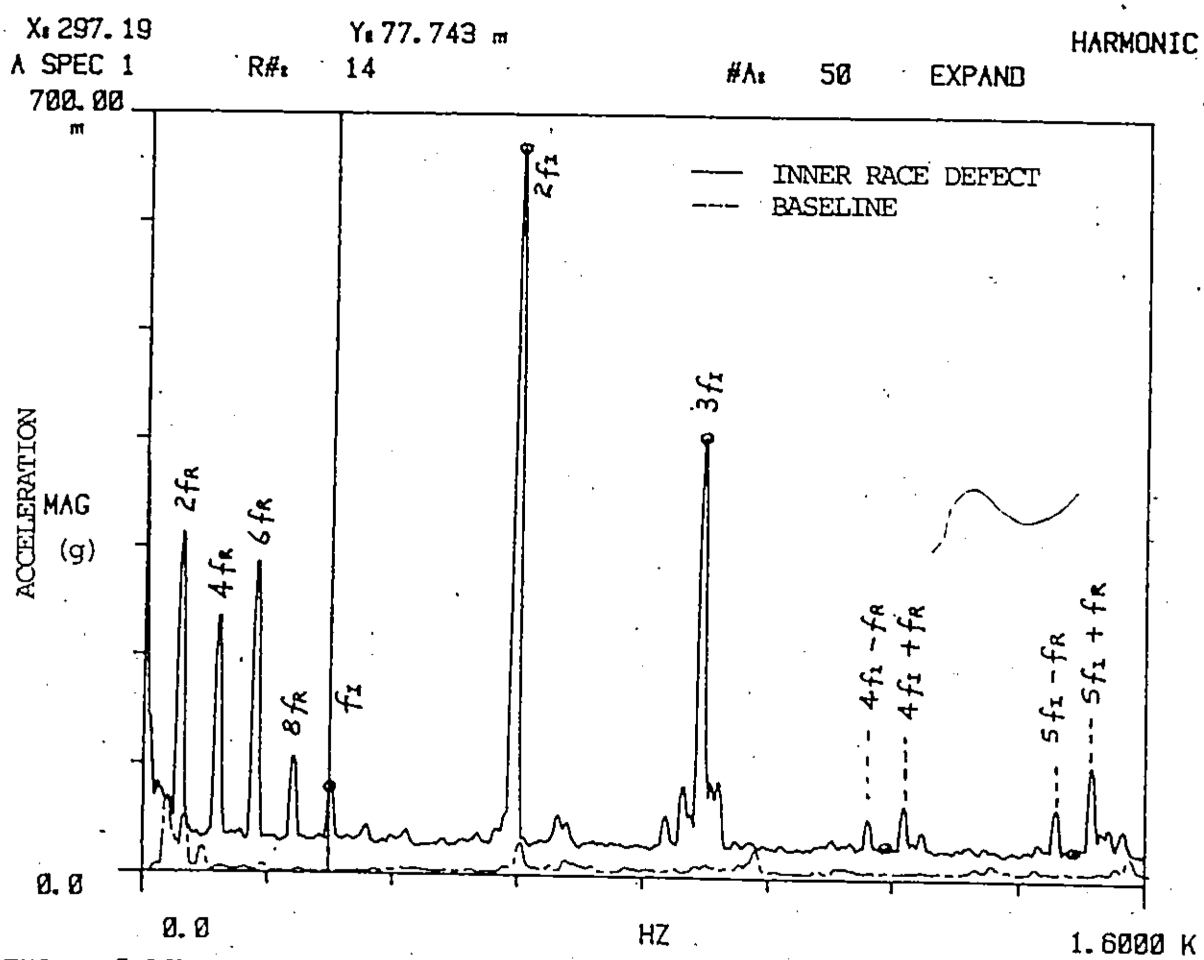


FIGURE 5.13b: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B, 0 - 1.6kHz

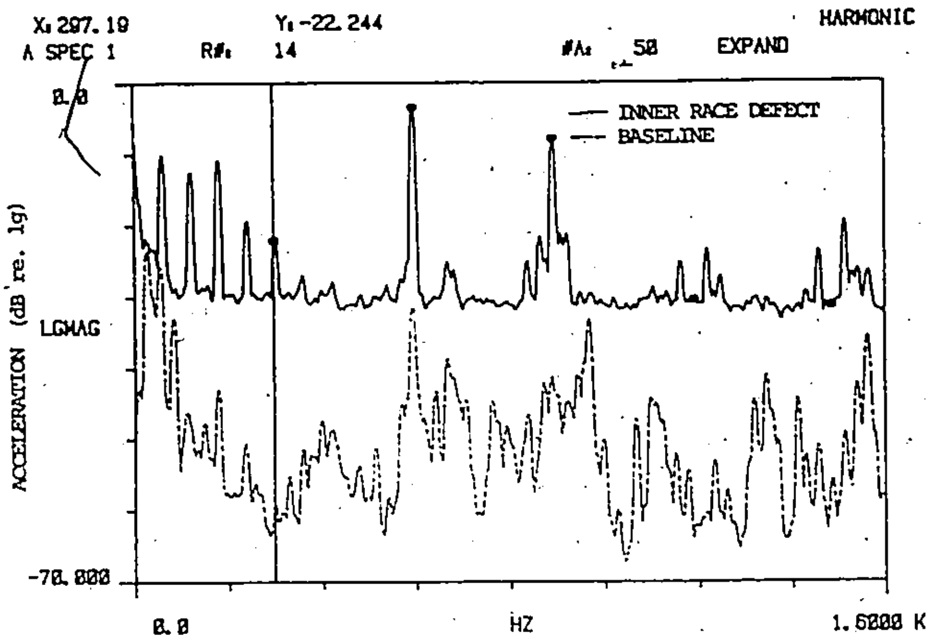


FIGURE 5.13c: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B (DECIBEL SCALE), 0 - 1.6kHz

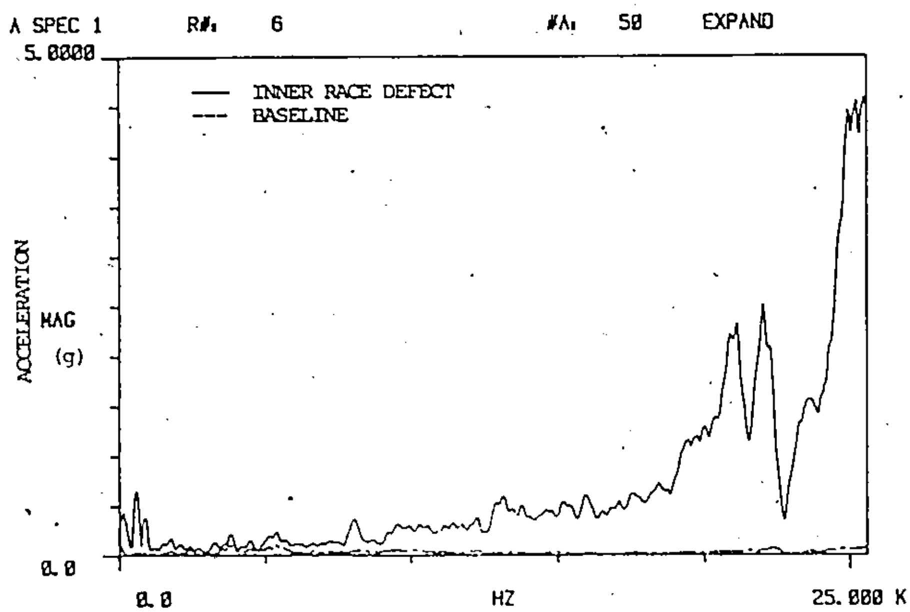


FIGURE 5.13d: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B, 0 - 25.6kHz

Furthermore, high frequency analysis (see Figure 5.13d) clearly illustrates the significant changes in the amplitudes of the spectra, especially in the frequencies above 7.0 kHz, for a bearing with an inner race defect. Again, high frequency analysis was found to be a repeatable, sensitive and reliable method of confirming the presence of an inner race defect. However, the source of each of the spectral components was difficult to define due to the lack of frequency resolution.

The time domain analysis, Figure 5.13a for measuring position 6B, indicates that there are well defined "pulses" as the balls pass over the defected inner race. The period of the pulses is found to be 3.4 milliseconds and thus the ball pass frequency of the inner race is computed to be 294 Hz. The difference between this frequency and the theoretical calculated BPFI is approximately 1 percent.

The presence of well defined "pulses" with constant period and hence frequency, is consistent for all the measuring positions.

Thus it is clear that time domain analysis is also reliable in identifying an induced inner race defect.

5.5.5 The Effects of Induced Multiple Defects

The frequencies generated by the induced multiple defects bearings and by rotating units generally add and substract. This phenomenon occurs frequently: thus some frequency spectra of multiple defected bearings will not only contain one of the five basic frequencies, f_R, f_A, f_B, f_I and f_O, but also the sum and difference of these basic frequencies.

The frequency spectra of induced multiple defects bearing is illustrated in Figure 5.14b with amplitude in linear scale and Figure 5.14c with amplitude in decibel scale. Some of the five basic frequencies can be identified but the sum and difference frequencies are almost always present in the spectra.

Frequency analysis of multiple defects bearing is very difficult because of its complex nature. The best approach is to determine any basic frequencies such as the five frequencies mentioned earlier and then the remaining frequency peaks are identified by trial and error.

In Figure 5.14b, all the frequencies generated by induced defects in the outer race, one ball and the inner race were identified and compared to the baseline spectrum. These include the five basic frequencies and also their sums and differences. The changes in amplitudes of the spectral

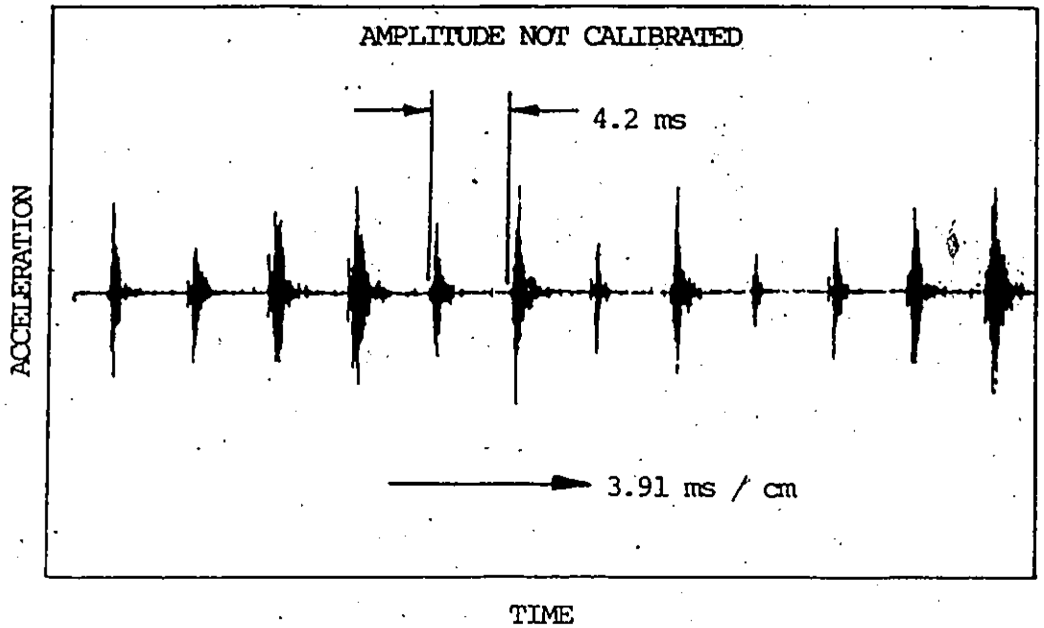


FIGURE 5.14a: ACCELERATION VERSUS TIME FOR MULTIPLE DEFECTS (POSITION 9B)

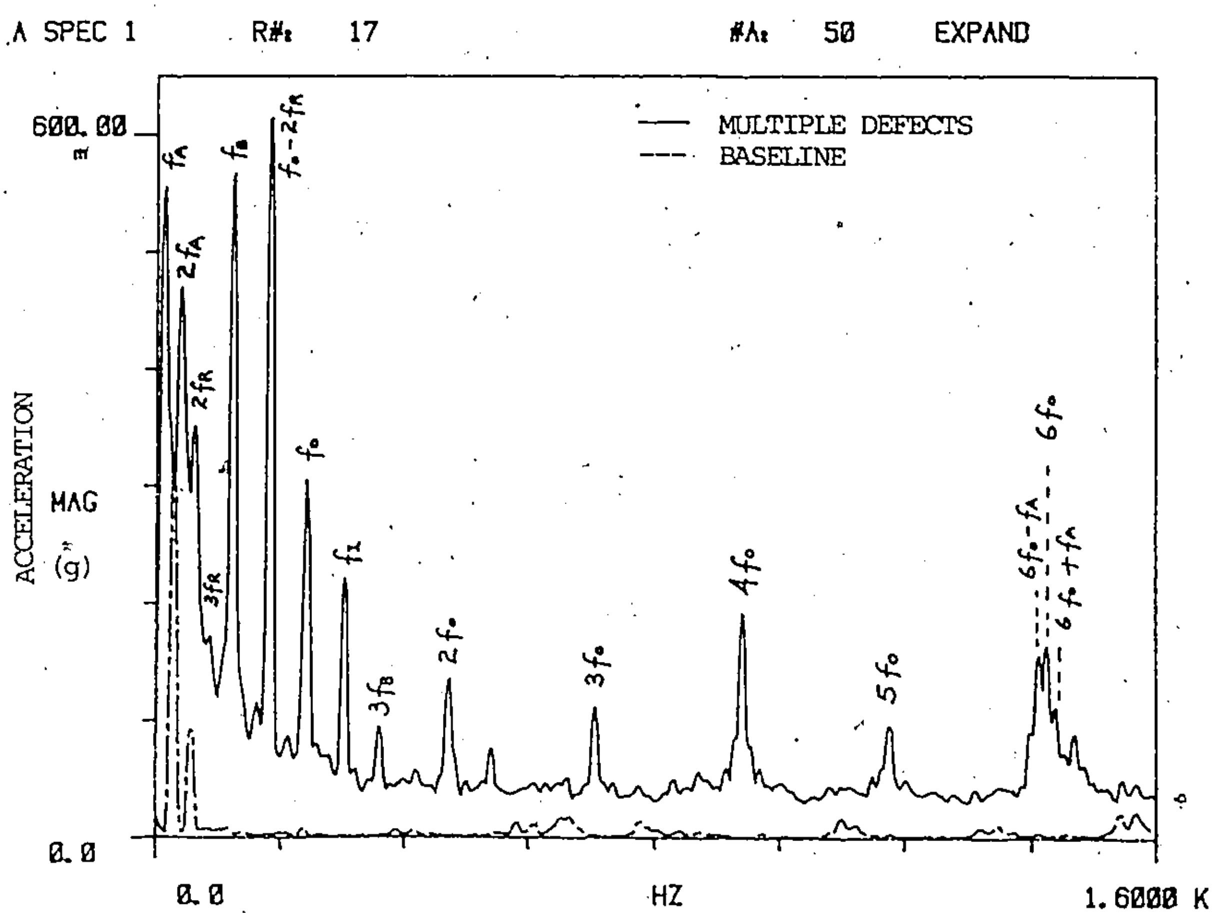


FIGURE 5.14b: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 9B, 0 - 1.6kHz

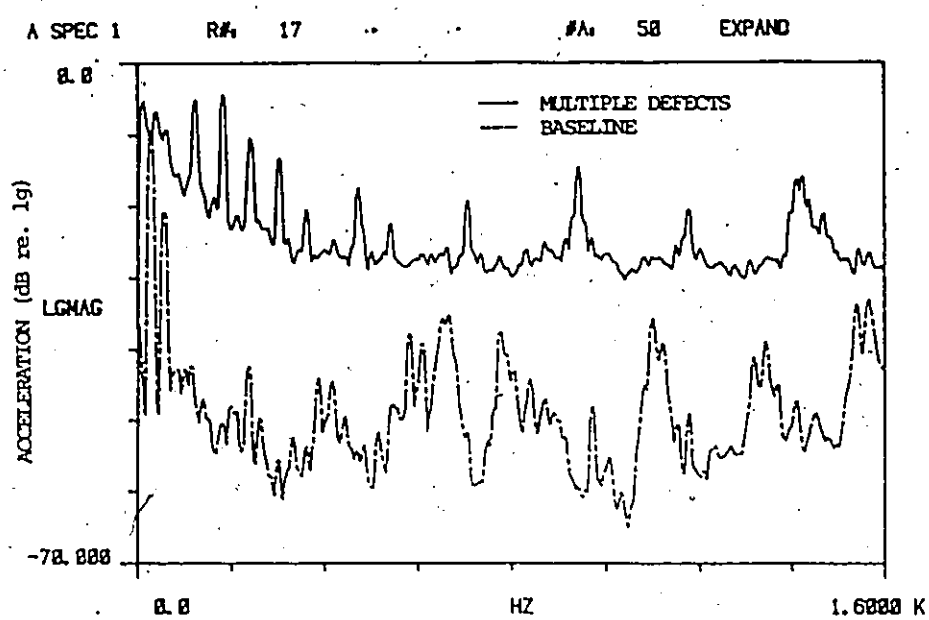


FIGURE 5.14c: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 9B (DECIBEL SCALE), 0 - 1.6kHz

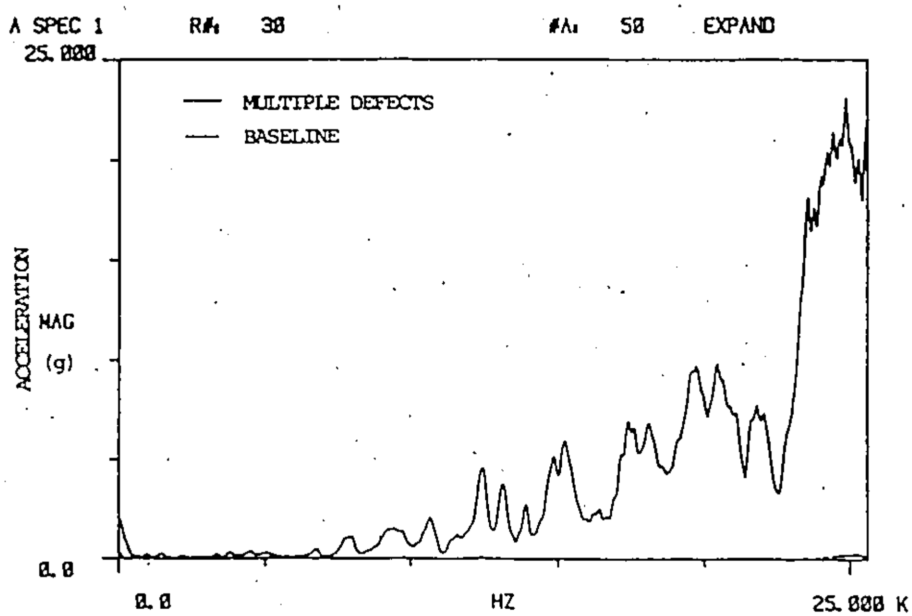


FIGURE 5.14d: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 9B, 0 - 25.6kHz

components of bearing with multiple defects and the baseline spectra vary among the measuring positions (see Appendix G).

The time domain data of the induced multiple defects bearing is generally more complex than that for bearings with one defect. Generally, it consists of pulses at frequencies corresponding to each particular defect. Some of these pulses were spaced closely together which made it more complicated to analyze.

VI. RECOMMENDED AREAS OF FUTURE RESEARCH AND DEVELOPMENT

6.1 EVALUATION OF BEARING CONDITION MONITORING

It has been mentioned in the literature survey that overall acceleration, velocity or displacement is generally inadequate for predicting bearing failure. This it is recommended that a study be undertaken to evaluate other bearing condition monitoring techniques, such as the shock pulse method, cepstrum, kurtosis, acoustic emission, etc., and possibly outline the problems associated with the application of these techniques to monitor typical bearings of the transfer machines.

6.2 THE ECONOMICS AND BENEFITS OF USING VIBRATION MONITORING SYSTEMS ON TRANSFER MACHINES

The major savings resulting from the application of vibration monitoring systems to transfer machines arise from the avoidance of production losses associated with the machine breakdown and the reduction in maintenance costs.

The cost / benefit analyses of vibration monitoring systems employed on transfer machines have not been documented in the literature.

Since the optimum vibration monitoring systems can only be selected if information is available on machinery repair costs and downtime costs, it is recommended that a study be undertaken with the cooperation of a typical user of transfer machine lines, to document such downtime and maintenance costs.

6.3 FUTURE VIBRATION MONITORING SYSTEMS

Future research is also recommended in the development of vibration monitoring systems which employ several analysis techniques and trending algorithms to monitor as well as diagnose the condition of transfer line machines. Furthermore, the system should also include the ability to correlate the vibration monitored with other process parameters and peripheral information (who was doing what, when). All of the above could be performed quite rapidly with the aid of a minicomputer.

VII. CONCLUSIONS

The following conclusions have been reached after examining the results of this study:

a. A survey of recently published literature indicates that the future trend is towards automated (computer based) vibration monitoring systems with fault diagnosis capabilities. A summary chart of the full bibliography with classification of technical papers by topics is given in Appendix B.

b. It has been shown that repeatable vibration measurements are possible under "in plant" conditions and that future trends in both the overall and spectral acceleration levels are readily apparent. Furthermore, for one particular machining station, accurate prediction of bearing failure is documented.

c. The induced defects (except a ball defect) on a bearing of a typical single spindle machining station can be identified by the frequency analysis as well as time domain analysis. Furthermore, it has been shown that incorrect installation of a bearing in a spindle assembly is also easily recognized using frequency analysis.

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APPENDIX A

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APPENDIX B

A SUMMARY CHART OF MACHINE HEALTH MONITORING BIBLIOGRAPHY

THOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
L.S.	1965	1	*							
la	1977	2		*		*			*	
.S.	1973	3	*	*	*	*	*			
.E.	1978	4		*	*					
	1983	5				*			*	*
	1976	6				*	*			
n, H.L.	1969	7				*	*			
.A.	1969	8	*			*	*			
, R.L.	1983	9				*			*	
, R.L.	1982	10							*	
, R.L.	1979	11	*			*				
, R.L.	1979	12				*				
	1974	13	*			*			*	
K.	1979	14				*	*		*	
.C.	1968	15	*			*				
.	1977	16				*				
, R.C.	1976	17				*	*			
v.d.	1977	18	*	*						
.J.	1977	19				*				
.J.	1976	20				*				
	1977	21				*				
P.	1964	22	*							
P.	1980	23				*			*	
P.	1978	24				*			*	
P.	1977	25				*			*	
P.	1977	26		*		*				
J.E.	1973	27		*		*				
J.E.	1972	28		*	*	*				
J.E.	1972	29	*	*	*	*				
.F.	1980	30							*	
P.	1983	31				*			*	*
te, K.G.	1981	32		*		*			*	
te, K.G.	1977	33							*	
	1979	34					*			
	1977	35				*				
	1977	36				*				
	1975	37				*				

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Braun, S.	1975	38				*				
Bredin, H.	1983	39		*	*					
Broch, J.T.	1980	40		*	*	*			*	
Bruel & Kjaer		41		*		*			*	
Bruel & Kjaer		42		*		*				
Bruel & Kjaer		43	*	*						
Bruel & Kjaer	1978	44		*						
Buckley, B.	1979	45							*	
Buehler, M.W.	1983	46	*			*				
Bultzo, C.	1976	47							*	*
Burchill, R.F.	1979	48				*				
Burchill, R.F.	1973	49				*	*			
Burke, E.C.	1982	50	*							
Burrows, C.R.	1981	51				*	*			
Canada, R.G.	1983	52				*			*	
Catlin, J.B.	1983	53				*	*			
Catlin, J.B.	1977	54		*	*					
Catlin, J.B.	1974	55				*				
Catlin, J.B.	1973	56				*				
Chapman, R.N.	1967	57				*			*	
Chen, R.P.	1973	58				*				
Chrisholm, R.	1976	59	*			*				
Chou, A.	1981	60							*	*
Clarke, D.	1980	61				*				
Cocheo, S.	1981	62							*	
Collacott, R.A.	1979	63	*	*	*	*	*	*	*	*
Collacott, R.A.	1976	64					*			
Collacott, R.A.	1976	65				*				
Collacott, R.A.	1975	66	*				*	*		
Darlow, M.S.	1975	67				*	*			
Dawson, B.	1976	68				*	*	*		
Dodd, V.R.	1982	69				*	*		*	*
Dodd, V.R.	1981	70				*	*		*	
Dodd, V.R.		71	*							
Donato, V.	1973	72		*						
Dornfeld, W.H.	1977	73	*			*				*
Dougherty, M.D.	1978	74				*			*	*

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Downham, E.	1971	75		•	•					
Dudgeon, E.H.	1978	76		•		•				
Dyer, D.	1978	77				•	•			
Dymac	1979	78							•	
Dymac	1979	79							•	
Edelman, S.	1978	80		•		•	•			
Edgerly, W.	1979	81				•				
Erskin, J.B.	1980	82				•			•	•
Erskin, J.B.	1980	83				•			•	•
Eshleman, R.L.	1983	84		•		•				
Eshleman, R.L.	1981	85	•	•	•	•				
Eshleman, R.L.		86				•				
Eshleman, R.L.	1976	87	•							
Filetti, E.G.	1977	88							•	
Finley, R.W.	1980	89				•				
Finley, H.F.	1978	90								
Fisher, L.L.	1980	91					•			
Flink, J.	1978	92				•				
Foster, G.B.	1967	93		•						
Fox, R.L.	1979	94				•	•			
Fox, R.L.	1977	95		•		•			•	
Fox, R.L.	1976	96	•	•		•				
Frarey, J.L.	1983	97		•						
Frarey, J.L.	1980	98		•						
Frarey, J.L.	1978	99				•			•	
Frarey, J.L.	1976	100		•		•				
Frarey, J.L.	1970	101				•				
George, P.T.	1978	102							•	
Glew, C.A.W.	1978	103				•			•	•
Glynn, T.J.	1978	104		•						
Gunning, L.C.	1980	105							•	•
Halloran, J.D.	1980	106				•				
Hammett, J.L.	1980	107							•	
Harbarger, W.B.	1976	108							•	
Harker, R.G.	1983	109							•	
Harker, R.G.	1977	110							•	

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Harrington, T.P.	1983	111							*	
Harris, C.H.	1976	112	*	*	*	*	*	*	*	*
Hauck, L.T.	1975	113							*	
Hegner, H.R.	1978	114							*	
Herzog, R.E.	1973	115				*				
Hewlett Packard	1981	116				*				
Hoffman, R.L.	1971	117			*	*				
Houser, D.R.	1978	118				*				
Houser, D.R.	1975	119				*				
Hudachek, R.J.	1976	120		*					*	*
Hulls, L.R.	1967	121				*				
Hvillum, J.J.	1967	122		*		*				
Jackson, C.	1979	123	*	*	*	*	*	*	*	*
Jackson, C.	1978	124				*				
Jackson, C.	1972	125		*	*					
Jackson, C.	1969	126		*	*					
James, R.	1976	127							*	
James, R.	1975	128		*	*					
James, R.	1973	129		*						
Kamperman, G.W.	1977	130		*						
Kaufman, A.B.	1976	131		*	*					
Kaufman, A.B.	1975	132		*	*					
Kaufman, A.B.	1975	133		*	*				*	
Keller, A.C.	1978	134		*	*	*				
Keller, A.C.	1975	135		*	*	*				
Kellum, G.B.	1973	136				*	*			
Kellum, G.B.	1972	137				*	*			
Kerfoot, R.E.	1974	138				*				
Kerfoot, R.E.	1973	139				*				
Kim, P.Y.	1983	140				*	*			
Kinne, H.H.	1969	141	*						*	
Kubiak, J.A.	1983	142				*				
Lang, G.F.	1976	143		*	*					
Lang, G.F.	1975	144		*	*					
Lavoie, F.J.	1969	145				*	*			
Libby, M.	1980	146				*			*	*

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Lorio, D.	1974	147		•					•	•
Ludwig, G.A.	1966	148	•							
Lundgaard, B.	1973	149	•							
Lura, L.E.	1972	150					•			
Lyon, P.H.	1983	151				•		•		
Maddox, V.	1977	152		•						
Maddox, V.	1973	153		•						
Martin, R.L.	1970	154					•			
Maten, S.	1970	155				•			•	
Maten, S.	1967	156			•					
Maxwell, S.A.	1982	157				•				
Maxwell, J.H.	1981	158	•			•				
Maxwell, J.H.	1980	159	•							
McElroy, J.W.	1983	160				•			•	
McGuckin, J.W.	1978	161				•				
McHugh, J.D.	1983	162		•	•					
McLain, D.A.	1981	163				•	•			
Meyer, L.D.	1980	164						•		
Mitchell, J.S.	1981	165	•	•	•	•	•	•	•	•
Mitchell, J.S.	1981	166							•	
Mitchell, J.S.	1980	167							•	
Mitchell, J.S.	1979	168		•						
Mitchell, J.S.	1977	169	•	•	•	•	•	•	•	•
Mitchell, J.S.	1972	170				•				
Mitchell, L.D.	1969	171				•				
Moller, H.M.	1982	172		•	•					
Moller, H.M.	1980	173					•			
Monk, R.G.	1972	174	•							
Morehead, J.C.	1983	175					•	•		
Morrow, R.S.	1975	176	•						•	
Myrick, S.T.	1982	177							•	
Neale, M.J.	1980	178							•	•
Nicolet, Scientific Corporation	1982	179	•	•	•	•	•	•		
Nicolet, Scientific Corporation	1981	180		•	•					
Nimitz, W.	1970	181		•	•					
Nishio, K.	1979	182		•			•			
Nittinger, R.H.	1970	183	•						•	

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Noon, R.	1979	184							*	
O'Doa, D.M.	1975	185		*					*	*
Palm, J.R.	1979	186				*				
Pandit, S.M.	1979	187				*				
Parmakian, J.	1962	188	*							
Patenaude, A.	1982	189							*	
Pekrul, P.J.	1976	190							*	
Pekrul, P.J.	1975	191							*	
Philips, G.J.	1980	192		*			*			
Philips, G.J.	1978	193		*			*			
Philips, G.J.	1975	194		*			*			
Piety, K.	1982	195							*	
Piety, K.	1979	196		*		*				
Piety, K.	1978	197				*				
Plunkett, R.	1976	198		*						
PMC/BETA	1980	199	*							
Price, M.H.	1977	200							*	
Psichogios, T.	1974	201	*							
Randall, R.B.	1983	202				*			*	
Randall, R.B.	1983	203				*				
Rathbone, T.C.	1939	204	*							
Redding, J.H.	1980	205		*					*	
Rickert, B.M.	1979	206							*	
Roger, L.M.	1979	207				*				
St. John, D.L.	1979	208	*			*				
Salamone, D.J.	1981	209				*	*			
Sankar, T.S.	1972	210				*				
Sattler, T.A.	1979	211		*	*					
Schanzenbach, G.P.	1975	212		*						
Schanzenbach, G.P.	1972	213		*						
Schlereth, F.H.	1972	214				*				
Schuh, D.	1978	215				*				
Sala, U.	1978	216							*	
Shatoff, J.	1976	217	*							
Shea, J.M.	1972	218				*				
Shea, J.M.	1969	219							*	

AUTHOR	DATE	REF. NO.	SOURCES OF VIBRATION	INSTRUMENTATION	MEASUREMENT TECHNIQUES	DATA PROCESSING TECHNIQUES	BEARING FAULTS ANALYSIS	GEAR FAULTS ANALYSIS	VIBRATION MONITORING SYSTEMS	USERS' EXPERIENCE
Simmons, P.E.	1978	220	*							
Smiley, R.G.	1983	221							*	
Smith, J.W.	1982	222							*	
Sohre, J.S.	1981	223	*							
Sparks, C.R.	1973	224				*				
Stephenson, B.J.	1979	225		*						
Sternlight, B.	1968	226	*							
Steward, R.M.	1981	227				*				
Strickler, R.A.		228	*							
Structural Kinematics Inc.	1983	229		*	*	*				
Structural Measurement System	1982	230							*	
Sundt, P.C.		231		*	*					
Sunnersjo, C.S.	1983	232							*	
Taylor, J.I.	1980	233				*	*			
Taylor, J.I.	1980	234				*	*			
Taylor, J.I.	1979	235				*	*	*		
Taylor, J.I.	1979	236				*		*		
Taylor, J.I.	1979	237				*	*			
Taylor, J.I.	1979	238				*	*			
Taylor, R.R.	1982	239		*					*	
Tustin, W.	1971	240	*	*	*					
Tustin, W.	1970	241		*						
Tustin, W.	1969	242	*							
Underwood, A.C.	1971	243	*							
University N.B.	1980	244	*	*	*	*	*	*	*	
Volin, R.H.	1979	245				*				
Welling, M.	1974	246		*					*	*
Wilson, D.S.	1975	247				*				
Wilson, J.	1978	248		*						
Winn, L.W.	1976	249					*			
Winston, G.C.	1971	250	*			*				
Witzer, K.Z.	1981	251		*						
Woptipka, J.L.	1971	252				*				
Wu, S.M.	1980	253				*				
Xistris, G.D.	1980	254				*				
Xistris, G.D.	1973	255				*				

APPENDIX C

BRIEF DESCRIPTIONS OF INSTRUMENTATION

BRIEF DESCRIPTIONS OF INSTRUMENTATION

1. Bruel and Kjaer Type 2209, Impulse Precision Sound Level Meter.

This instrument is used for measuring sound and vibration. It is equipped with an individually calibrated condenser microphone, an internal reference voltage for calibration, "Linear", "A", "B", "C" and "D" weighting networks, impulse and peak detector with "Hold" circuits as well as "Fast", "Slow" and "Impulse" meter responses, overload warning lamps and AC and DC output for connection to a level recorder. An interchengeable meter scale and a range attenuator scales facilitate the direct reading of vibration (as well as sound or voltage) over a wide range.

The frequency response of the condenser microphone (B & K 4165) is 4 Hz to 12.5 kHz (± 1 dB) while that of the amplifier is 2 Hz to 70 kHz (± 1 dB).

2. Bruel and Kjaer Type 2218, Precision Integrating Sound Level Meter.

This instrument is used for measuring both sound and vibration. It includes analog readout and an L_{eq} meter with digital readout. The rms detector has the standardized Impulse, Fast and Slow time constants and a Peak Hold feature with a 50 μsec rise time. The full measuring range of 25 to 145 dB is covered in three overlapping 80 dB ranges. AC and DC outputs are available for connection of recorders.

The frequency response of the microphone (B&K 4165) is 4 Hz to 12.5 kHz ($\pm$ 1 dB) while that of the amplifier is 20 Hz to 20 kHz ($\pm$ 1 dB).

3. <u>Bruel and Kjaer Type 2635, Charge Amplifier</u>

This unit is a portable, low noise charge amplifier intended for vibration measurements with a piezoelectric accelerometer input. It has an internal 160 Hz sinusoidal reference signal. The frequency response extends from 0.2 Hz to 100 kHz ($\pm$ 10%). The calibrated output ranges for acceleration are from 0.1 mV/ms^2 to 1 mV/ms^2.

4. <u>Bruel and Kjaer Type 4291, Accelerometer Calibrator</u>

This unit is a laboratory and field portable calibrator. It provides sinusoidal acceleration output of 1g peak at 79.6 Hz.

5. <u>Bruel and Kjaer Type 4333 Accelerometer</u>

This accelerometer uses piezoelectric material which when physically stressed, produces an electrical output. The main specifications are:

a. Voltage sensitivity = 18.8 mV/g
b. Charge sensitivity = 20.9 pC/g
c. Weight = 12.7 grams
d. Resonant frequency = 36 kHz

6. Bruel and Kjaer Type 4370 Accelerometer

This accelerometer also uses piezoelectric material which when physically stressed, produces an electrical output. The main specifications are:

a. Voltage sensitivity = 85.0 mV/g

b. Charge sensitivity = 99.5 pC/g

c. Weight = 53.9 grams

d. Resonant frequency = 17 kHz

7. Gould OS4020, Digital Storage Oscilloscope

This unit provides a combination of digitally stored and real time displays. The main specifications include:

Vertical Deflection

Two identical input channels

Bandwidth : DC - 10 MHz

Sensitivity : 5 mV/cm to 20 V/cm

Accuracy : ± 3%

Time Base

Range : 1 μs/cm to 20 s/cm in 23 ranges

Accuracy : ± 3%

8. Hewlett Packard 5423A, Structural Dynamics Analyzer

This is a two channel Fast Fourier Transform Analyzer with built-in modal analysis package. Both time domain and frequency domain measurements can be performed. The related performance specifications are:

a. Frequency range: 0 to 25.6 kHz

b. Frequency resolution: 16 Hz to 100 Hz

c. Dynamic range: 75 dB

9. Hewlett Package 7045A, X-Y Recorder (Plotter)

This unit is a one-pen very high speed X-Y recorder. It is connected to the oscilloscope to produce the time domain data (amplitude versus time).

10. Hewlett Packard 9872B, Digital Graphic Plotter

This unit is a four-pen microprocessor based digital plotter which create hard copy graphics. It is connected to the Hewlett Packard 5423A, Structural Dynamics Analyzer to produce both tables and graphs.

11. Nagra IV-SJ, Reel to Reel Tape Recorder

This unit is a two channel direct record/reproduce reel to reel tape recorder. At 15 in/sec speed, the frequency response of this tape recorder extends from 25 Hz to 35 kHz.

12. Piezotronics Inc., PCB Model 303A02 Accelerometer

This model ICP (Integrated Circuit Piezoelectric) accelerometer is designed specifically for high frequency response. The miniature size of this accelerometer makes it ideal for vibration and shock measurements where low mass is important. The main specifications are:

a. Frequency range = 0.7 to 20,000 Hz (+ 10%)

b. Resonant frequency = 70 kHz

c. Voltage sensitivity = 9.47 mV/g

13. Piezotronics Inc., PCB Model 408B06, Power Supply

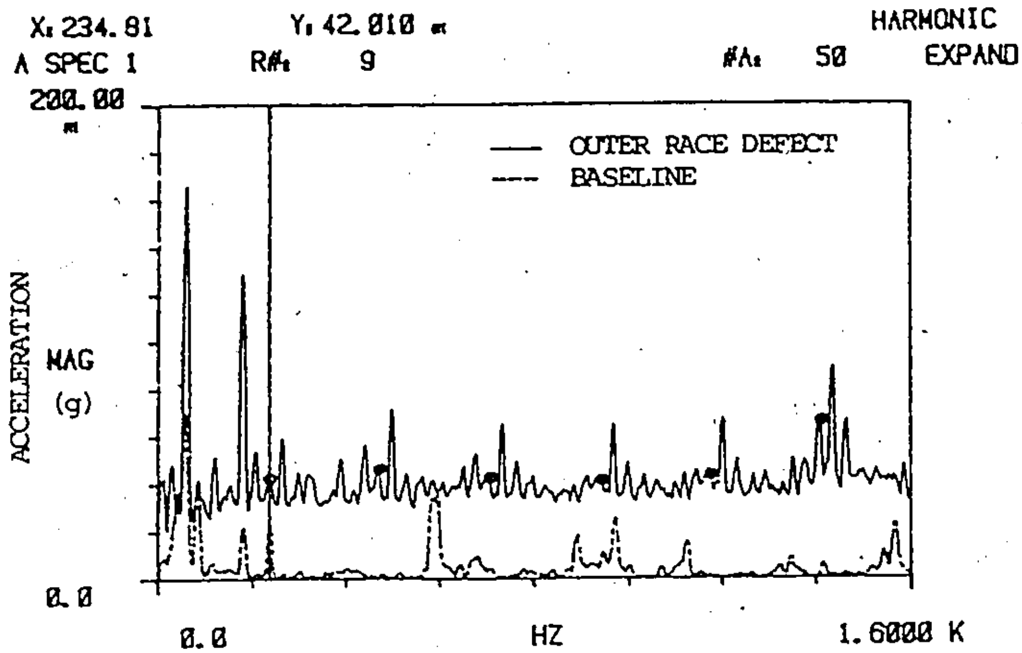
This power supply unit consists of a well regulated 18-27 VDC source, a current regulating diode and a method for decoupling the signal. It is for use with an ICP accelerometer.

14. TEAC R-61D, Cassette Data Recorder

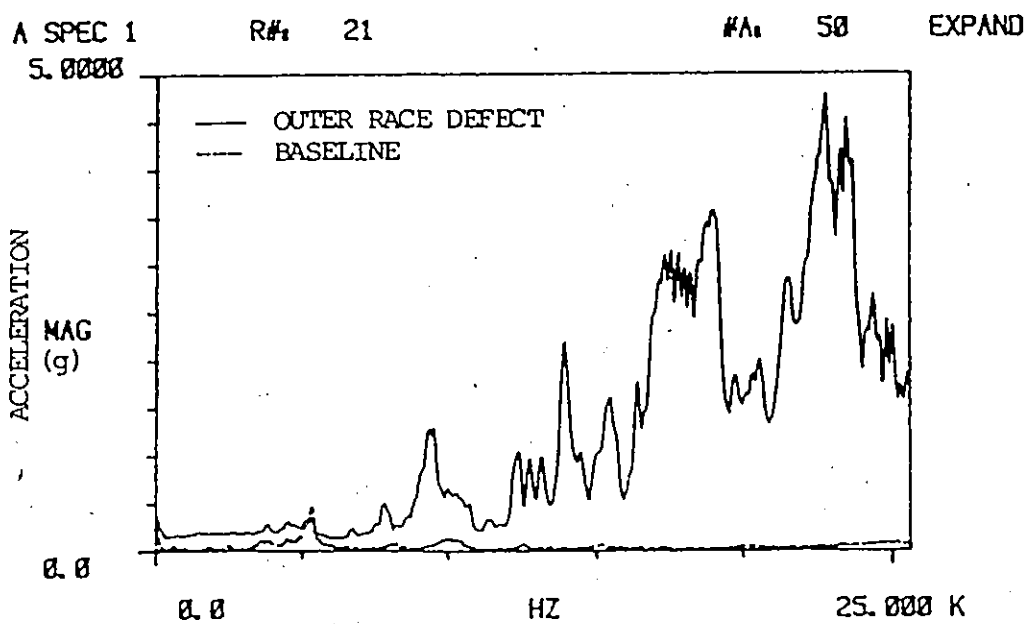
It is a four channel recorder: two FM record/reproduce amplifiers and four direct record/reproduce amplifiers. Analog signals of 50 Hz to 8000 Hz can be recorded and reproduced by direct mode, and those of DC up to 625 Hz by FM mode.

APPENDIX D

FREQUENCY SPECTRA FOR INDUCED OUTER RACE DEFECT BEARING

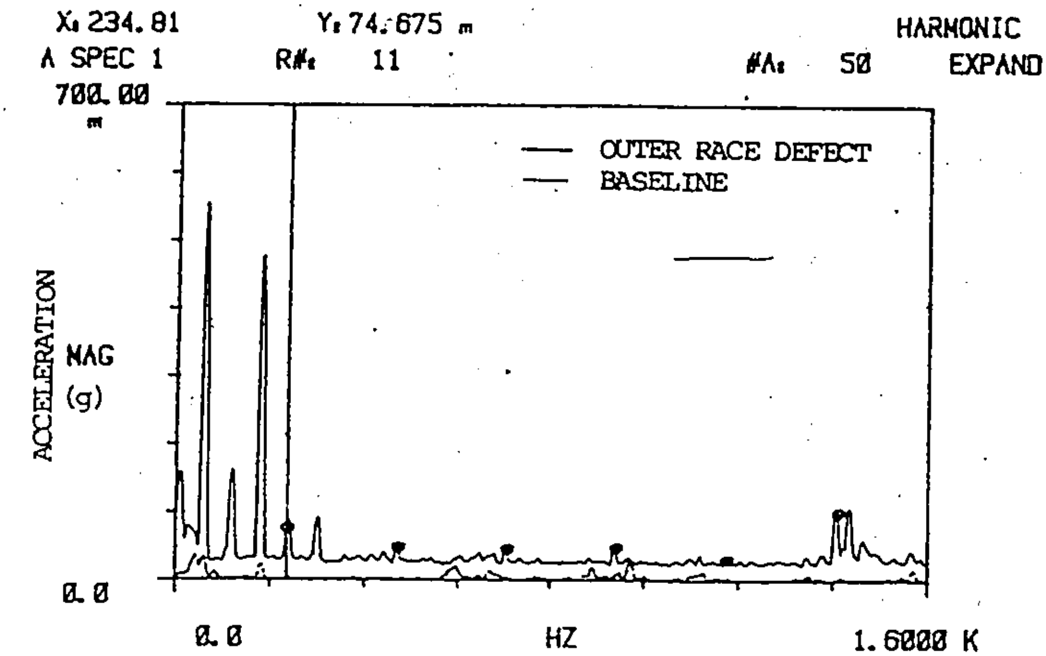


(a)

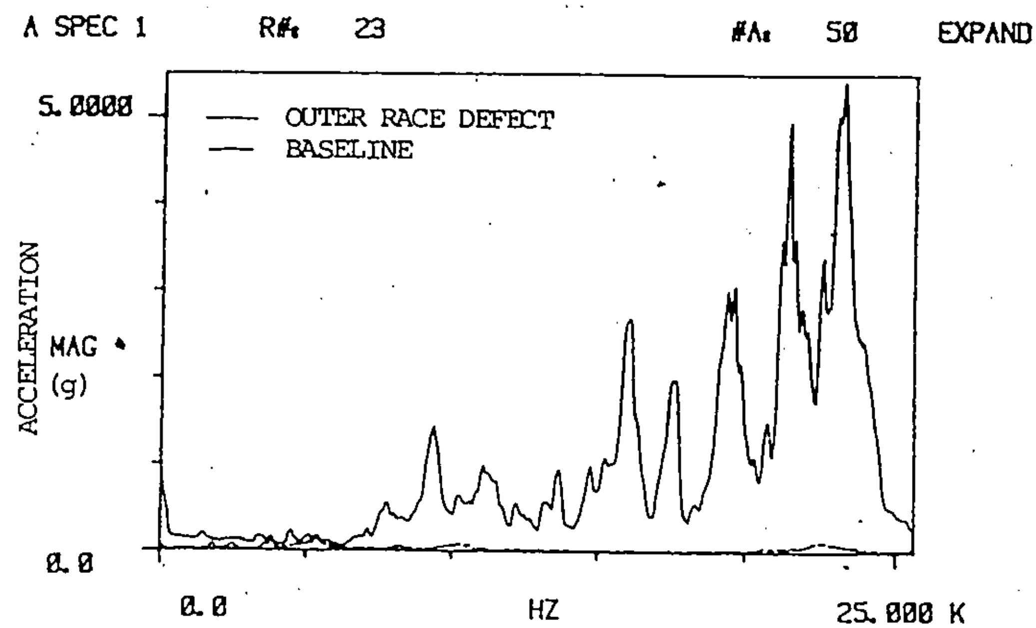


(b)

FIGURE D1: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 1P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

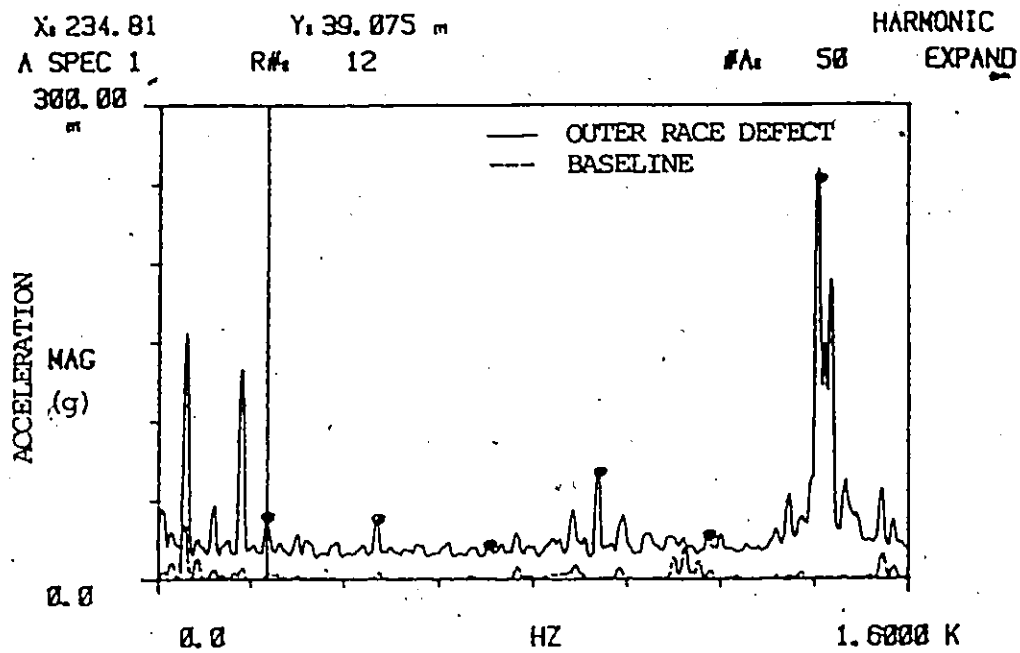


(a)

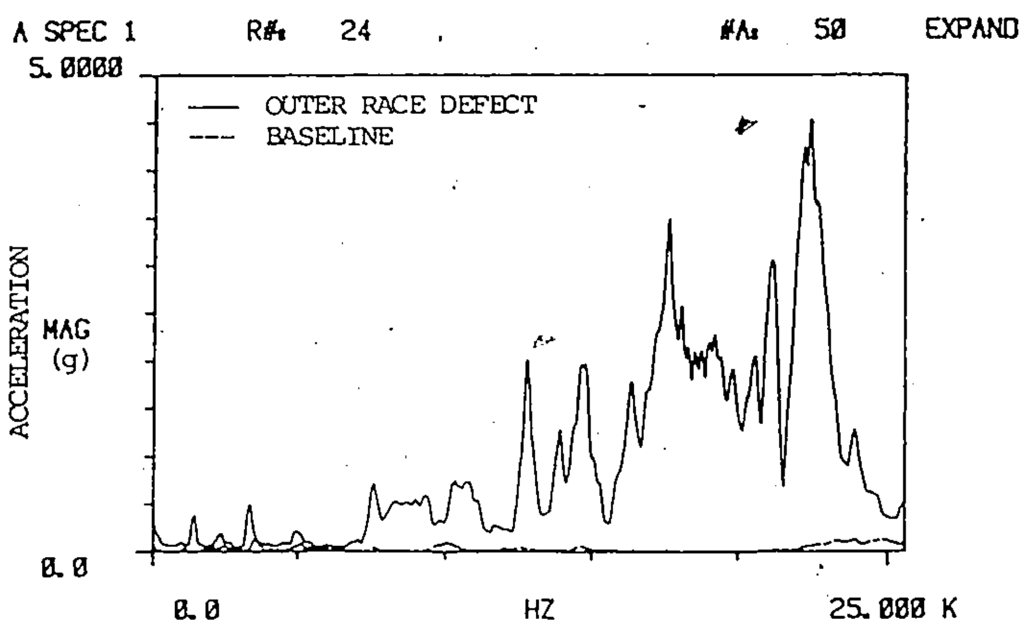


(b)

FIGURE D2: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 2P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE D3: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 3P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

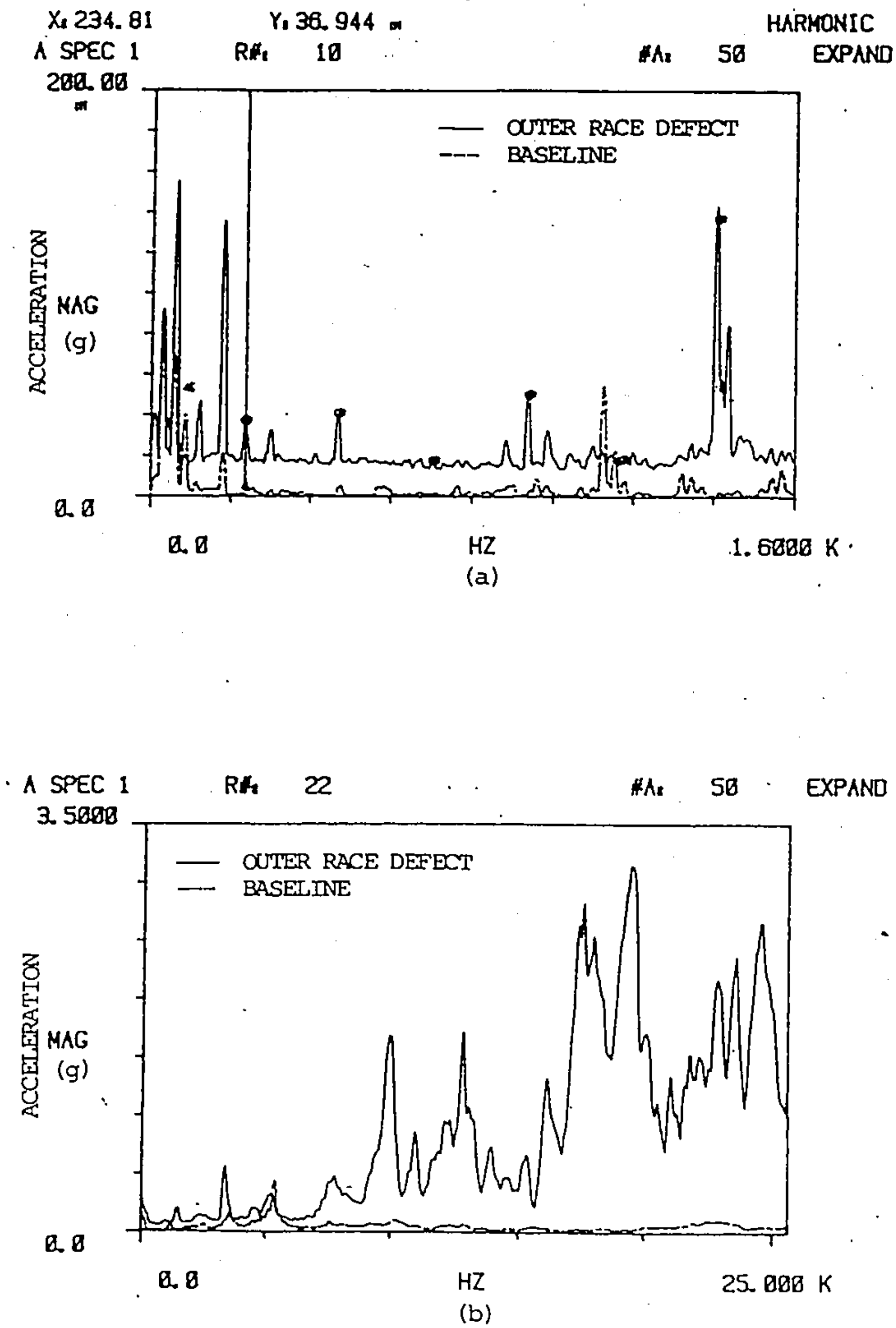
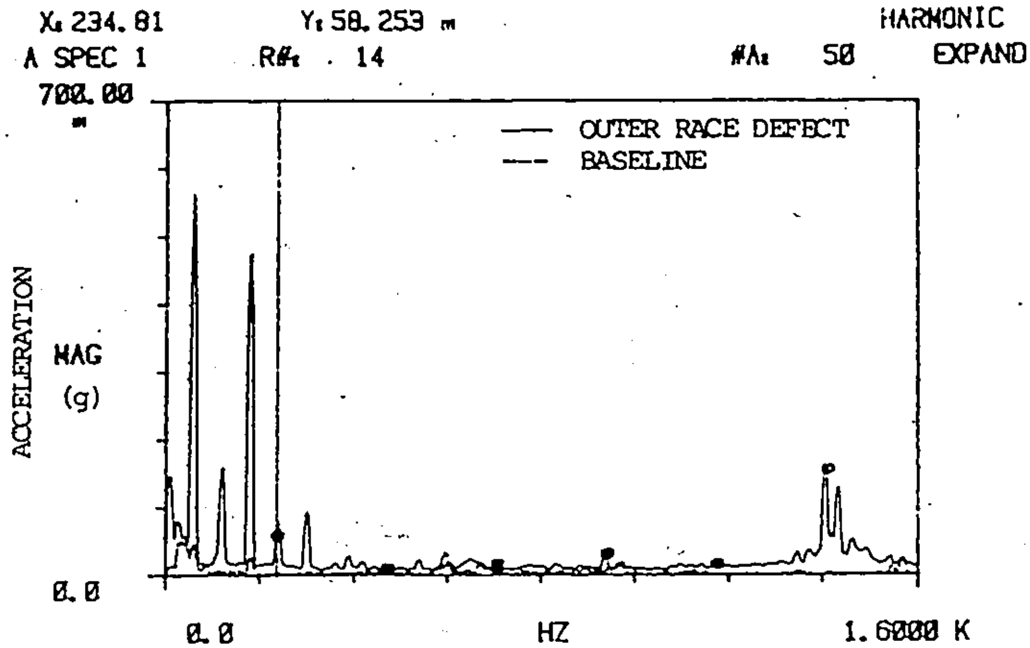
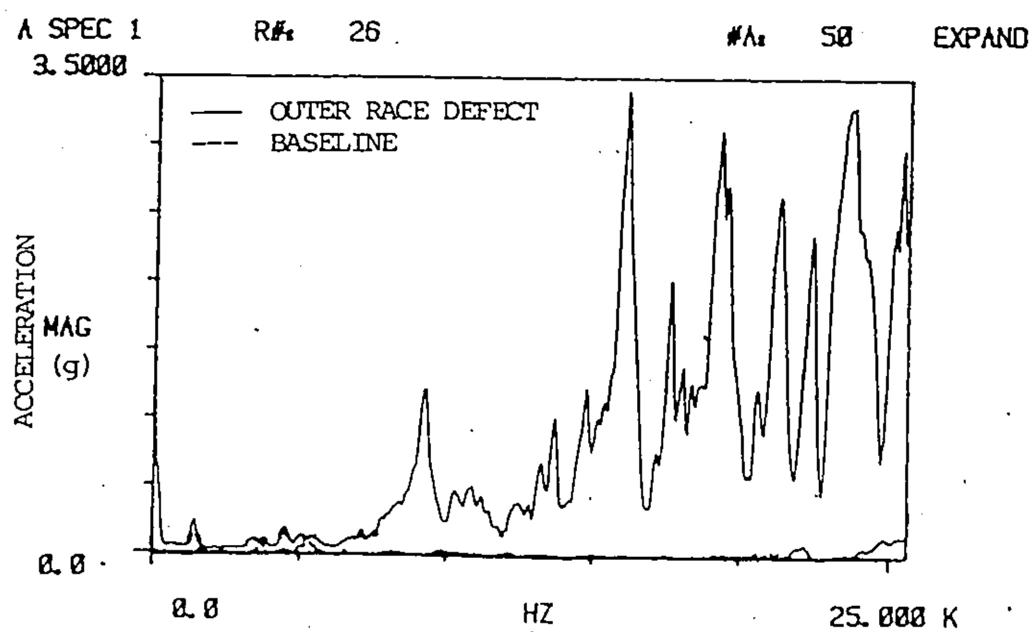


FIGURE D4: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 4P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE D5: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

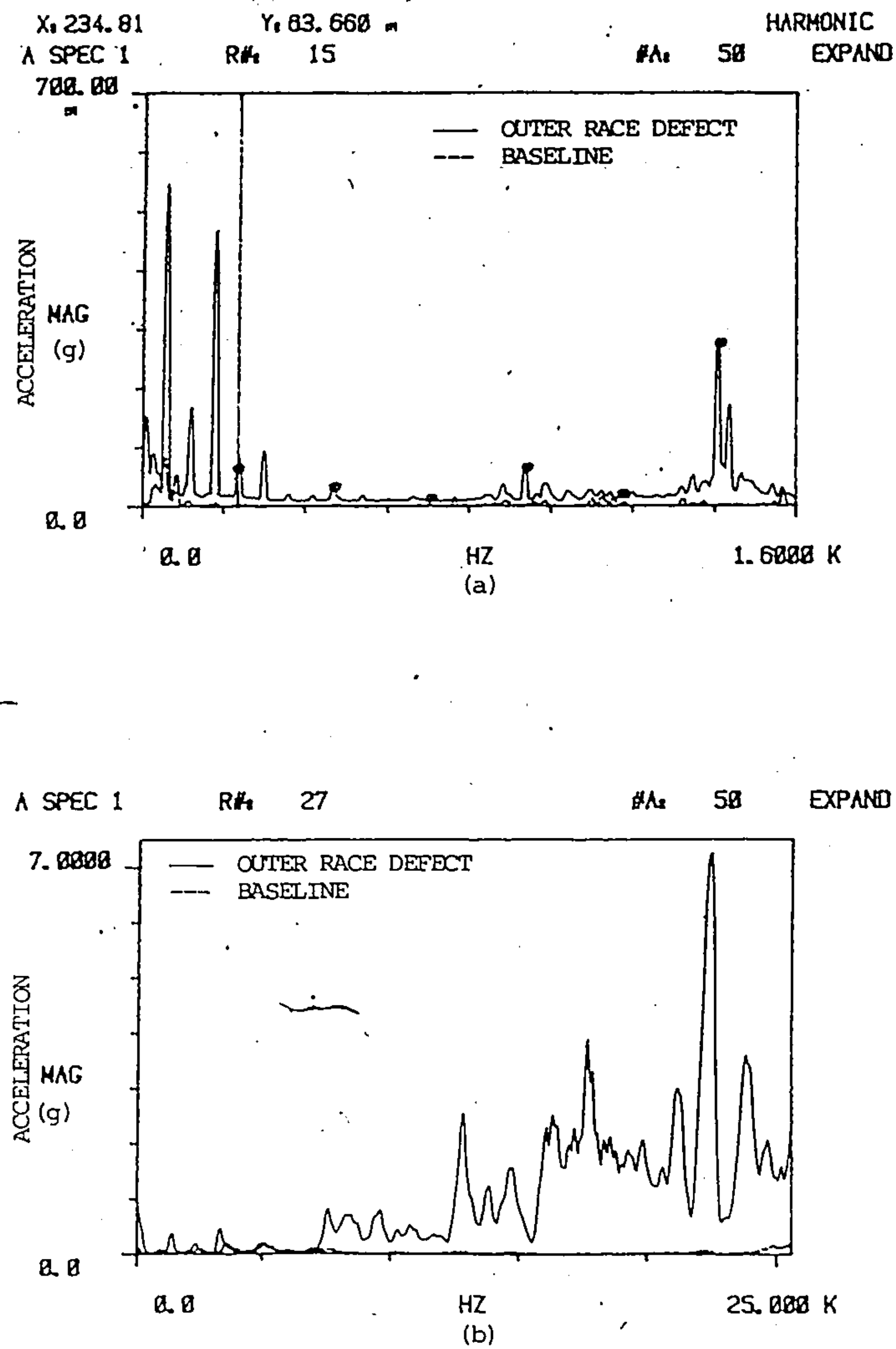
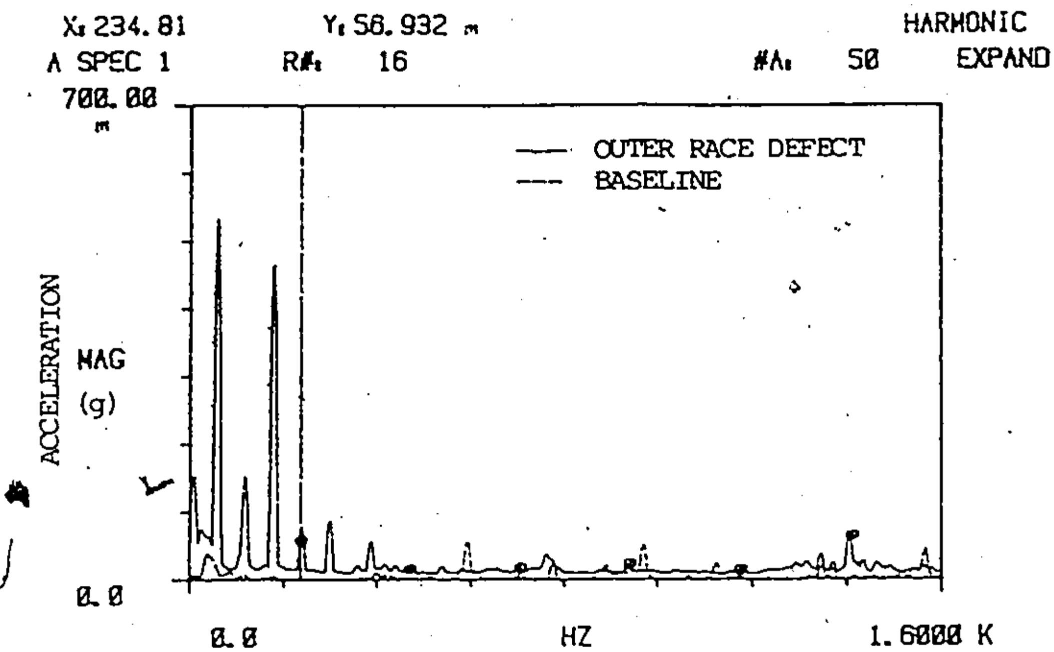
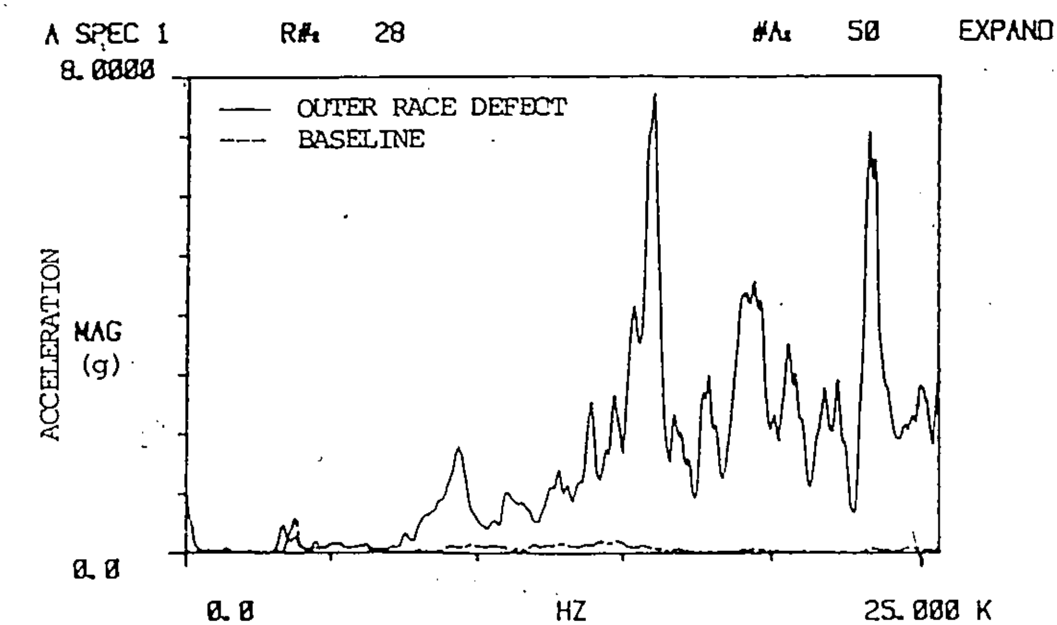


FIGURE D6: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 7B (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE D7: OUTER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 8B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

APPENDIX E

FREQUENCY SPECTRA FOR INDUCED BALL DEFECT BEARING

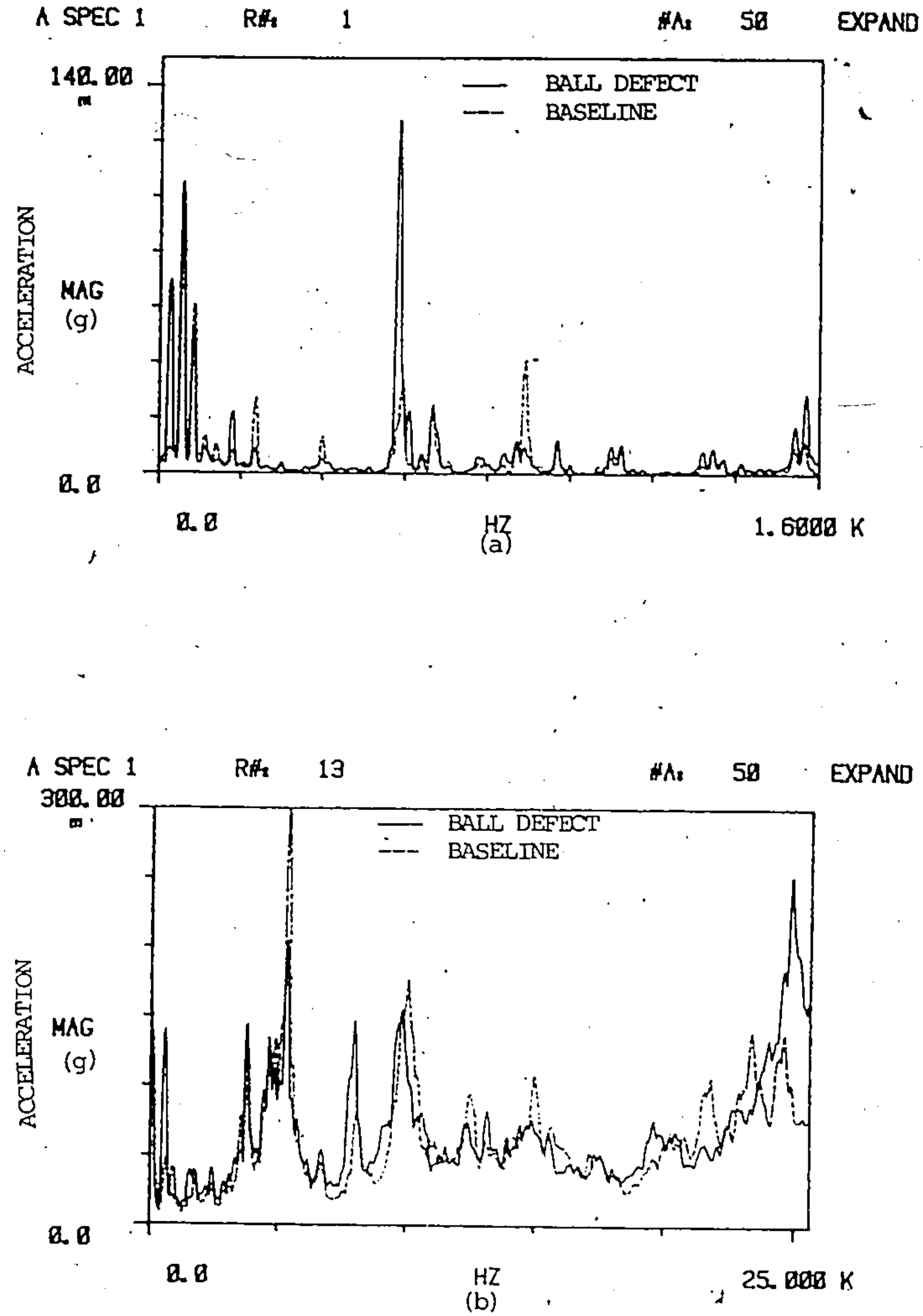
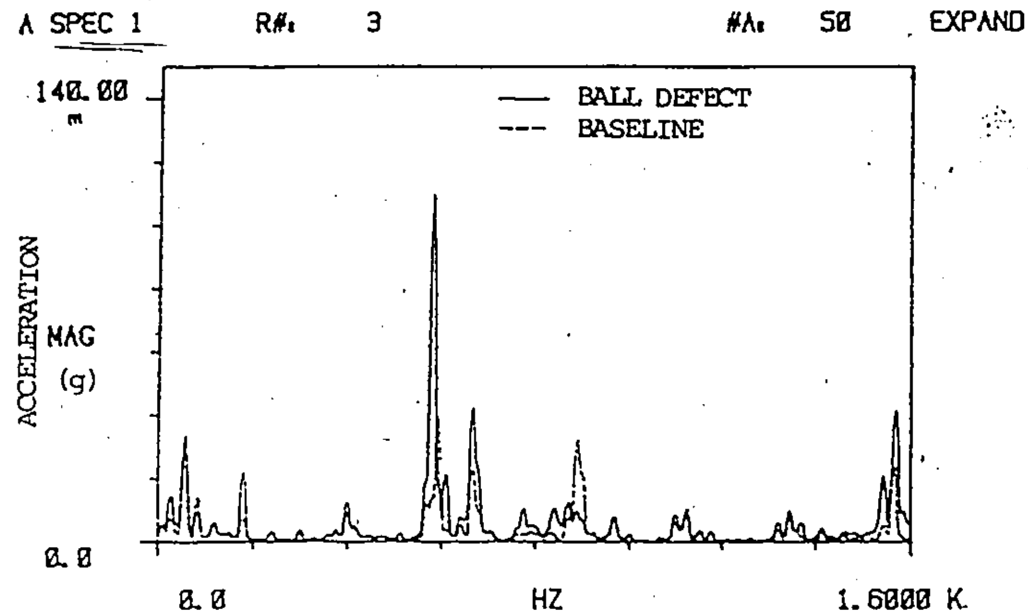
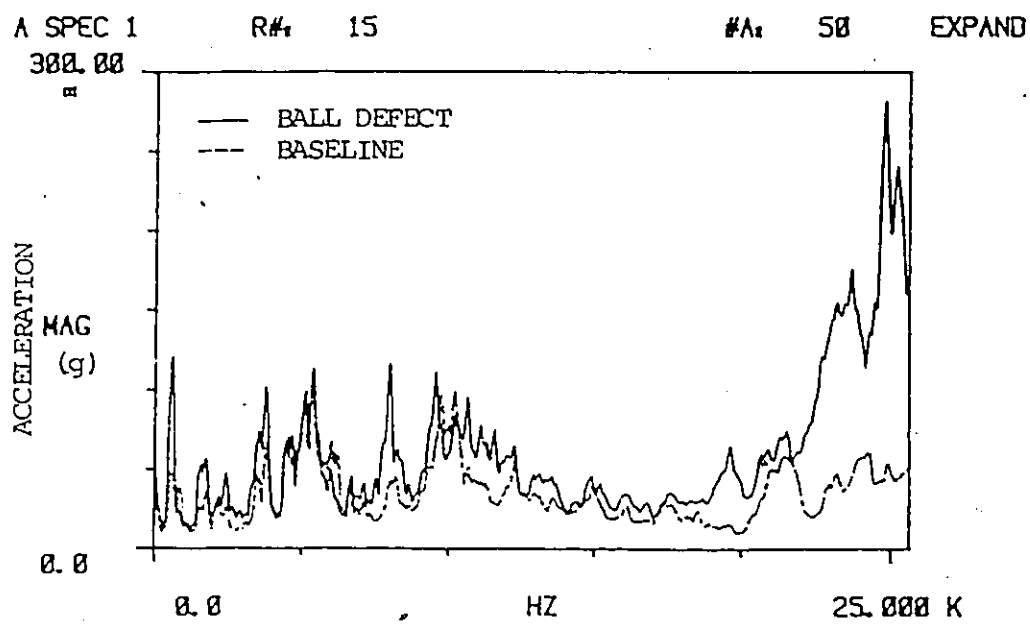


FIGURE E1: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 1P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

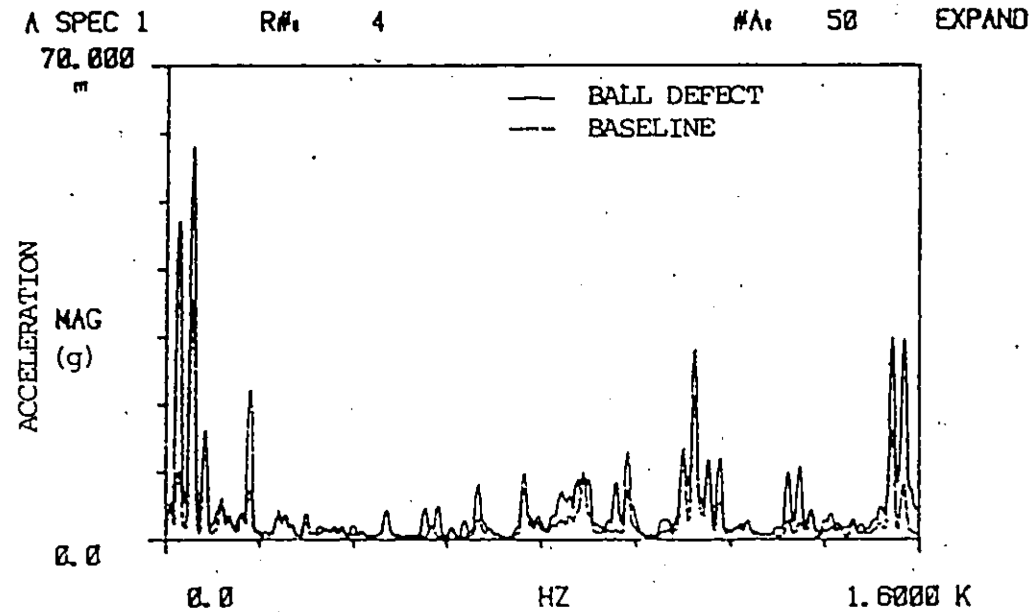


(a)

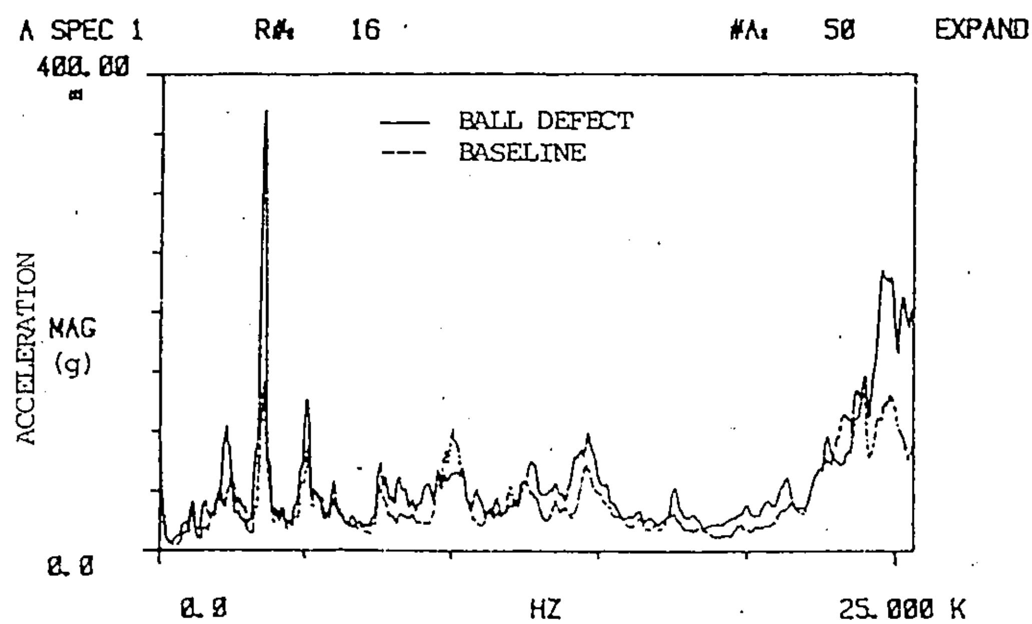


(b)

FIGURE E2: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 2P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

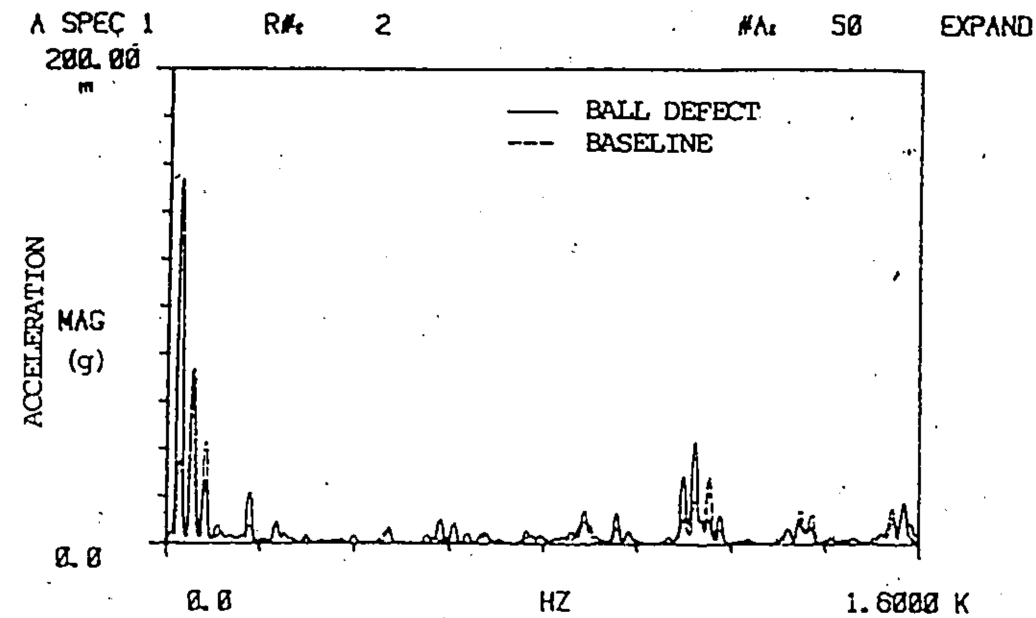


(a)

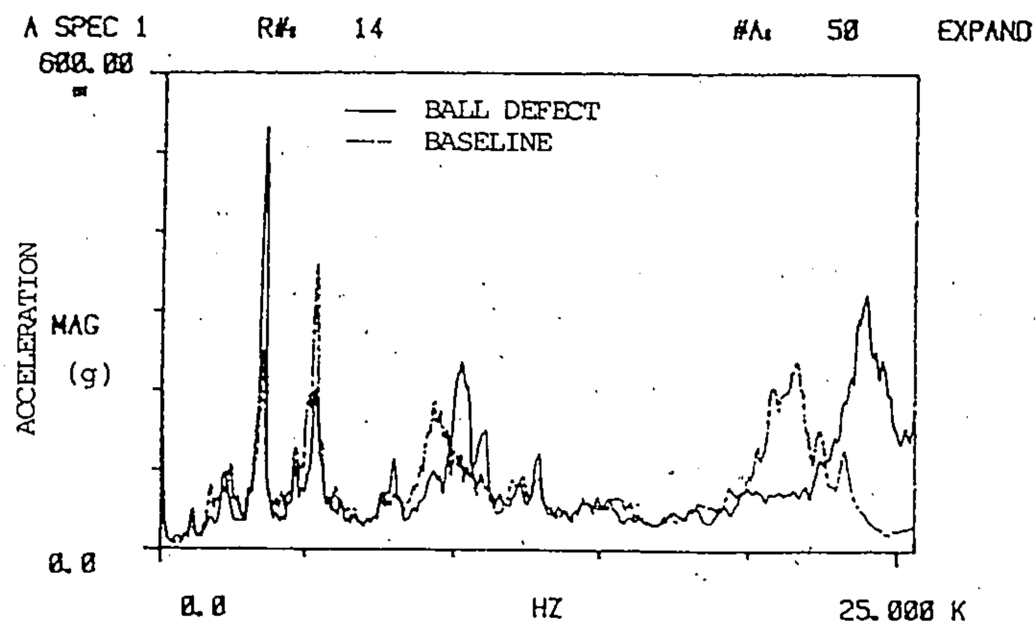


(b)

FIGURE E3: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 3P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

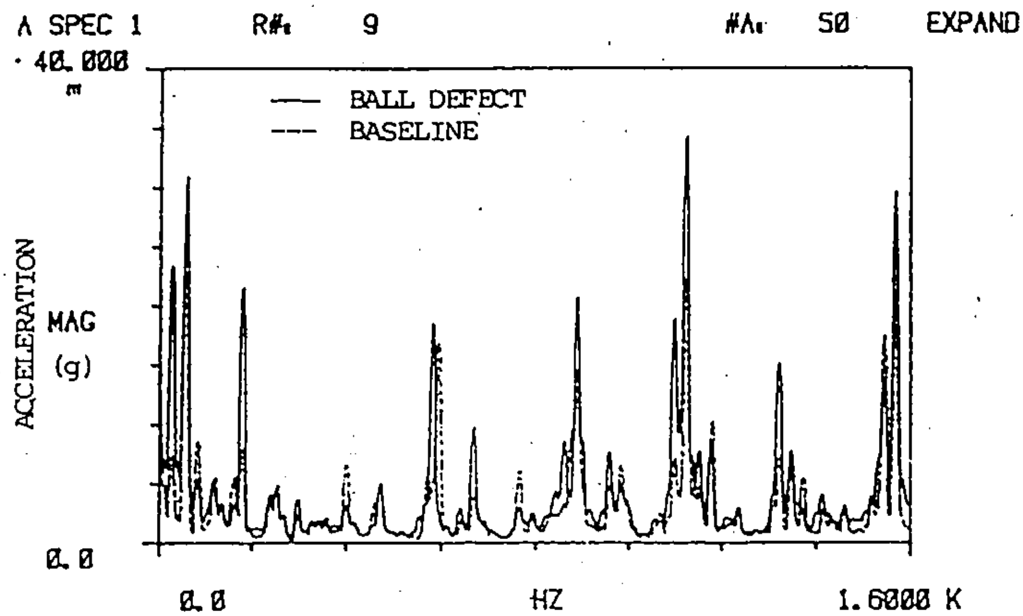


(a)

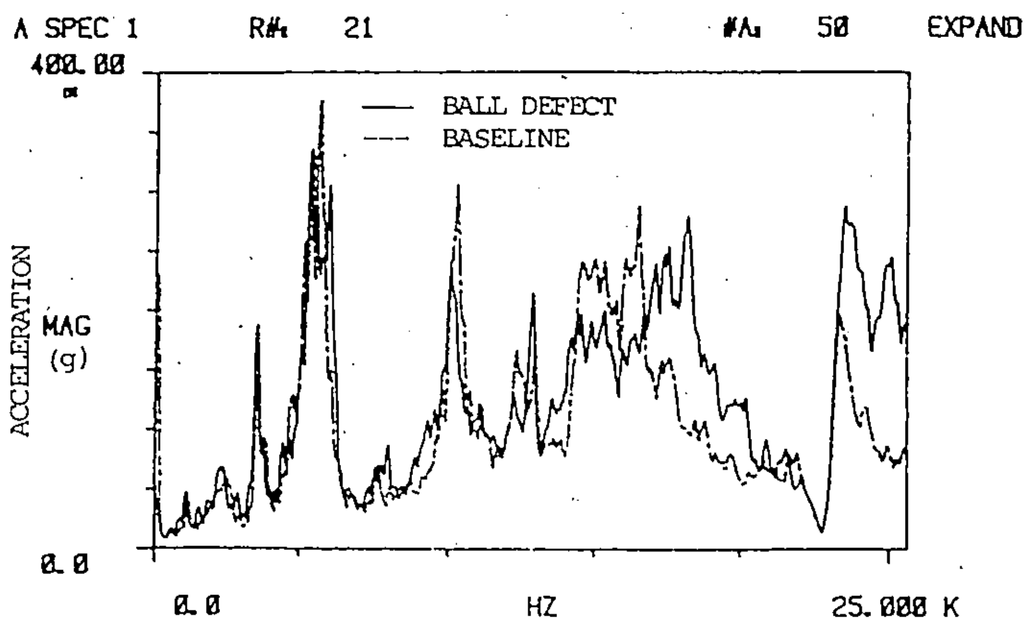


(b)

FIGURE E4: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 4P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE E5: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

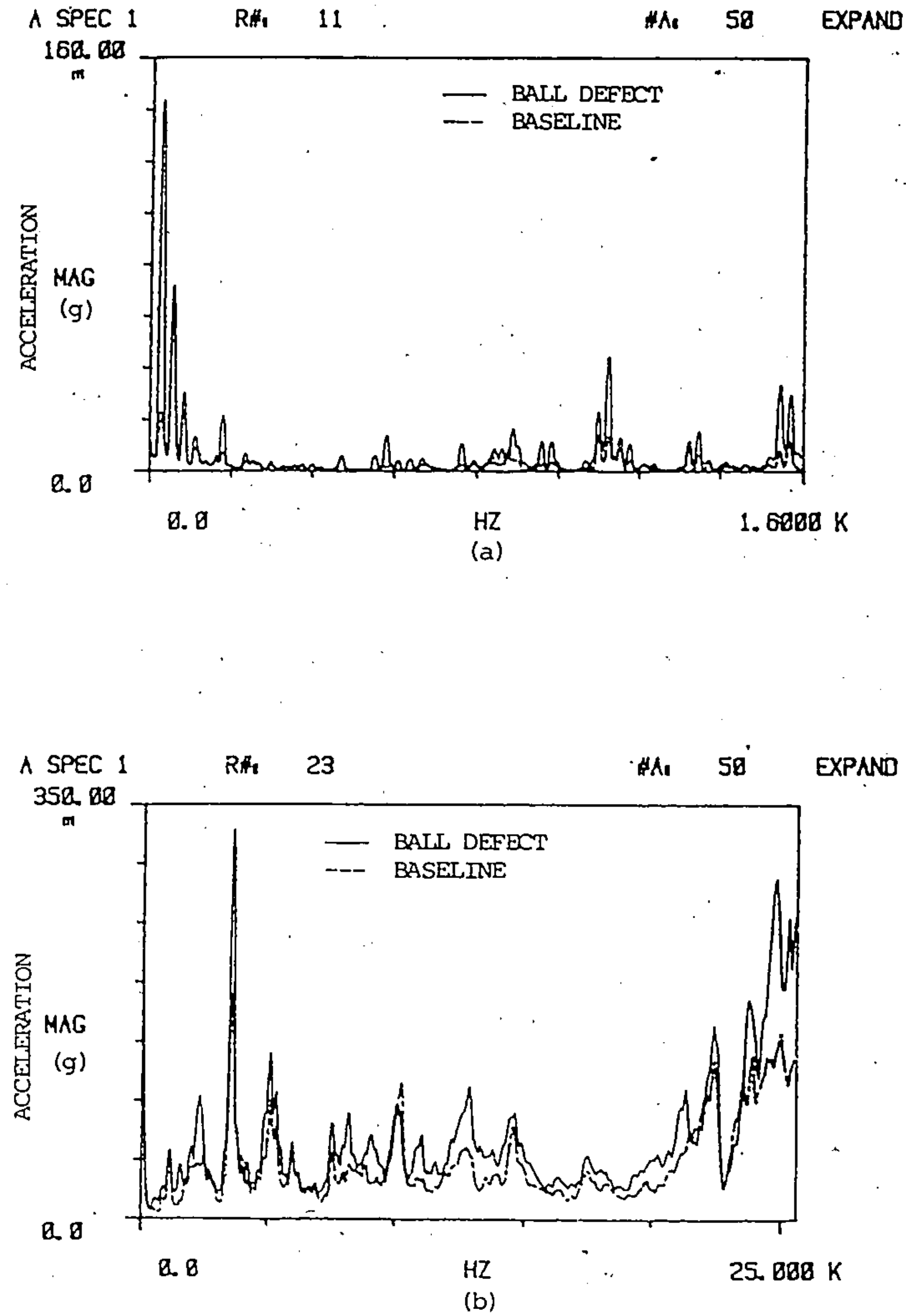
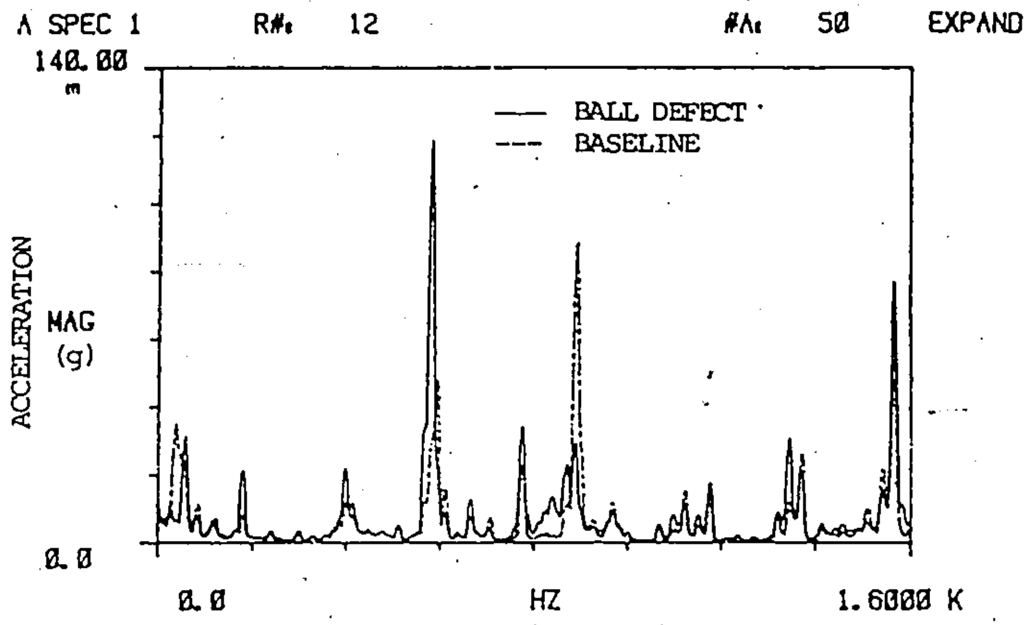
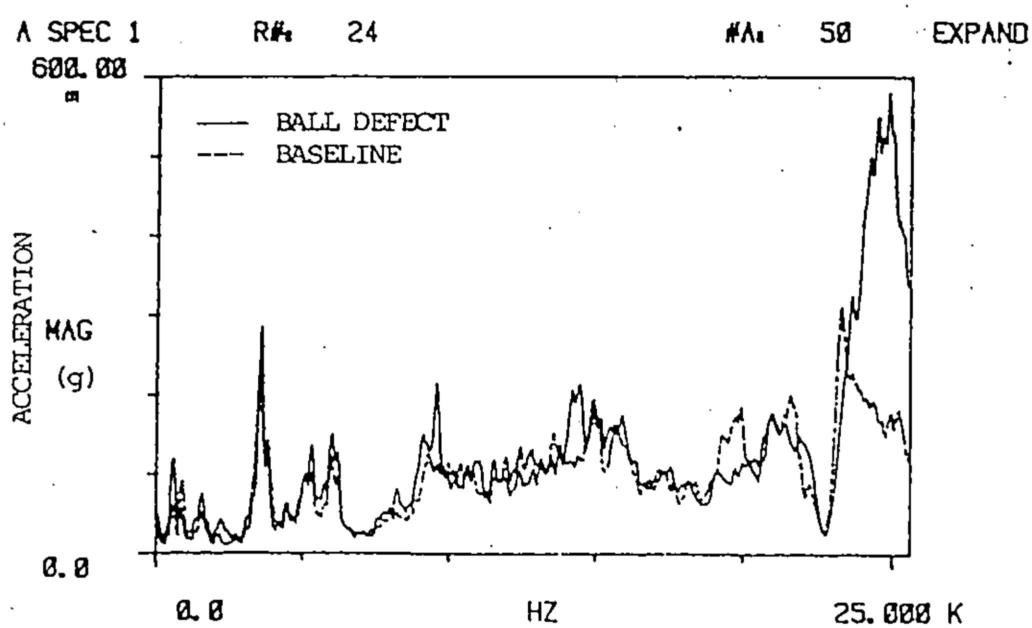


FIGURE E6: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 7B (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)

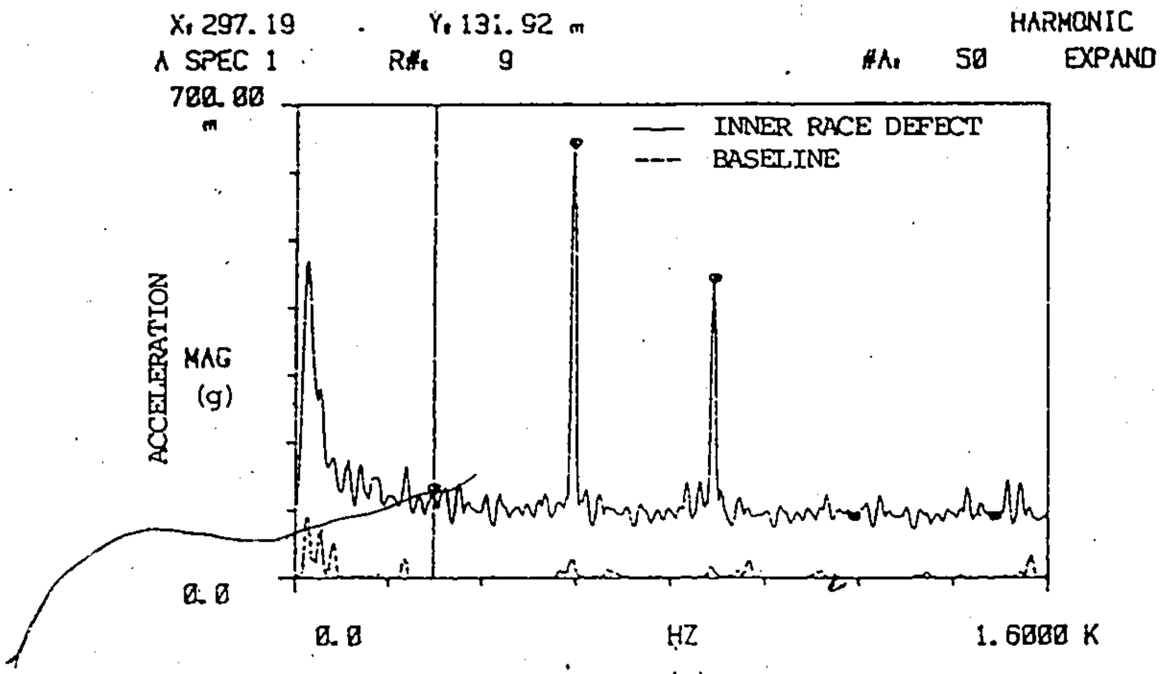


(b)

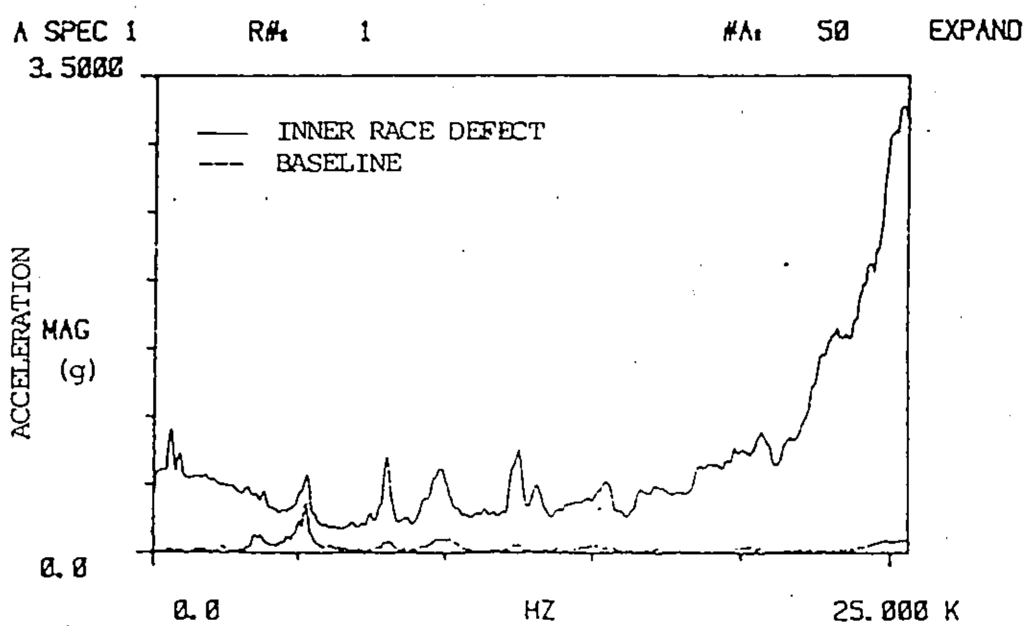
FIGURE E7: BALL DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 8B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

APPENDIX F

FREQUENCY SPECTRA FOR INDUCED INNER RACE DEFECT BEARING



(a)



(b)

FIGURE F1: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 1P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

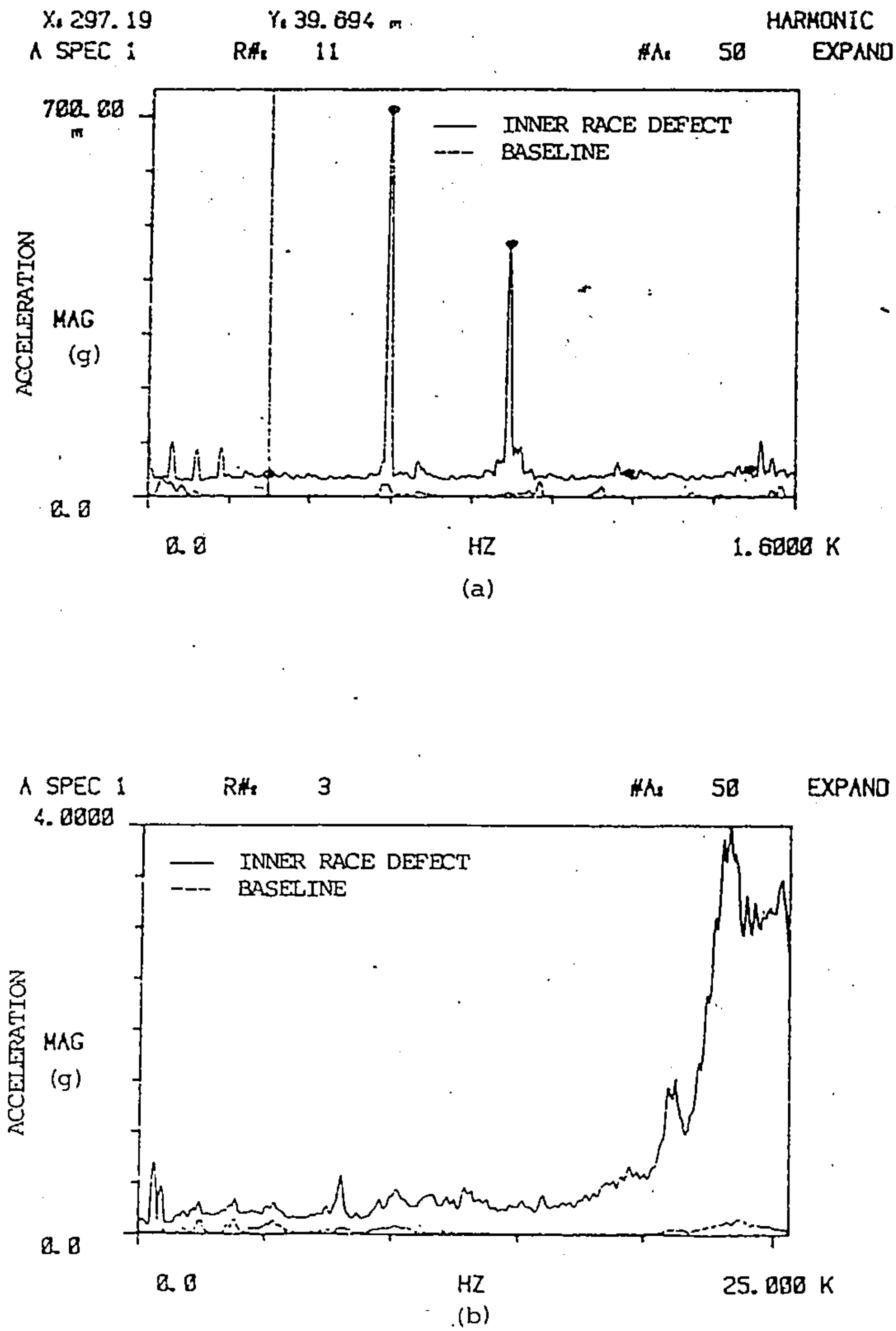
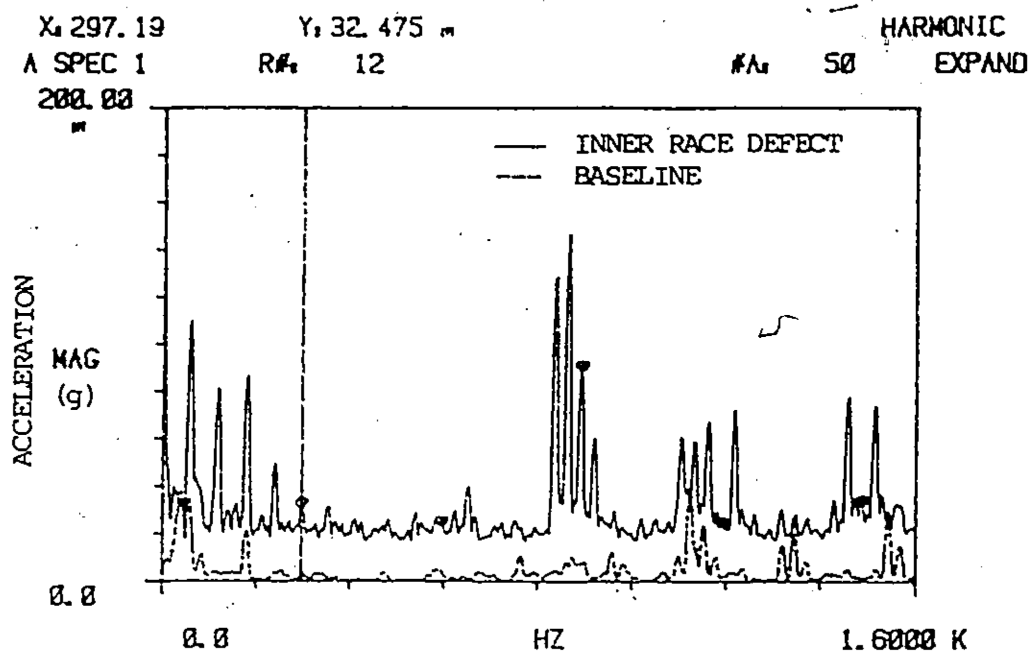
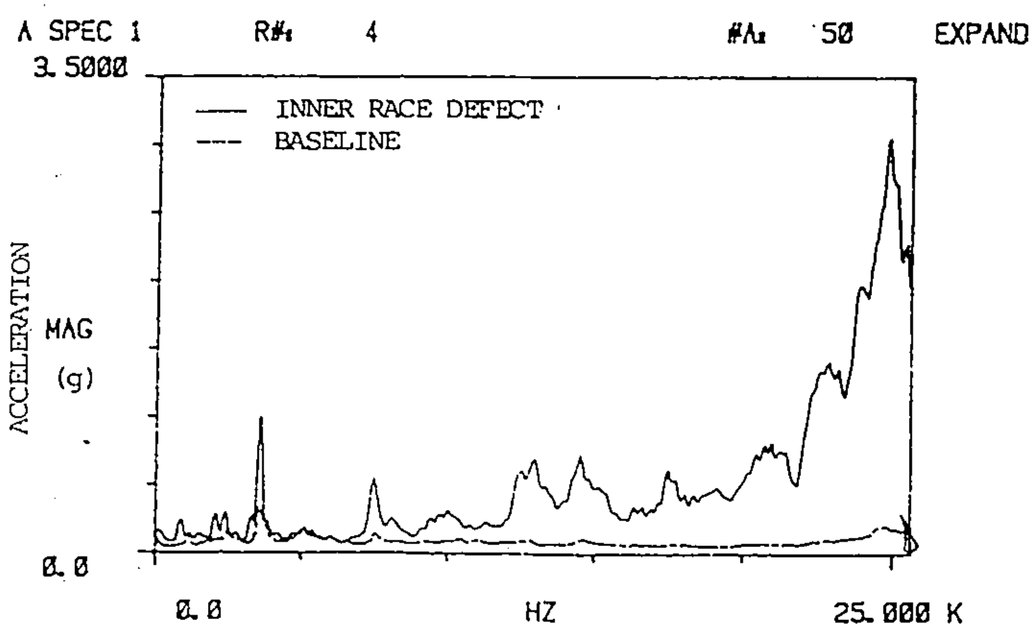


FIGURE F2: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 2P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE F3: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 3P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

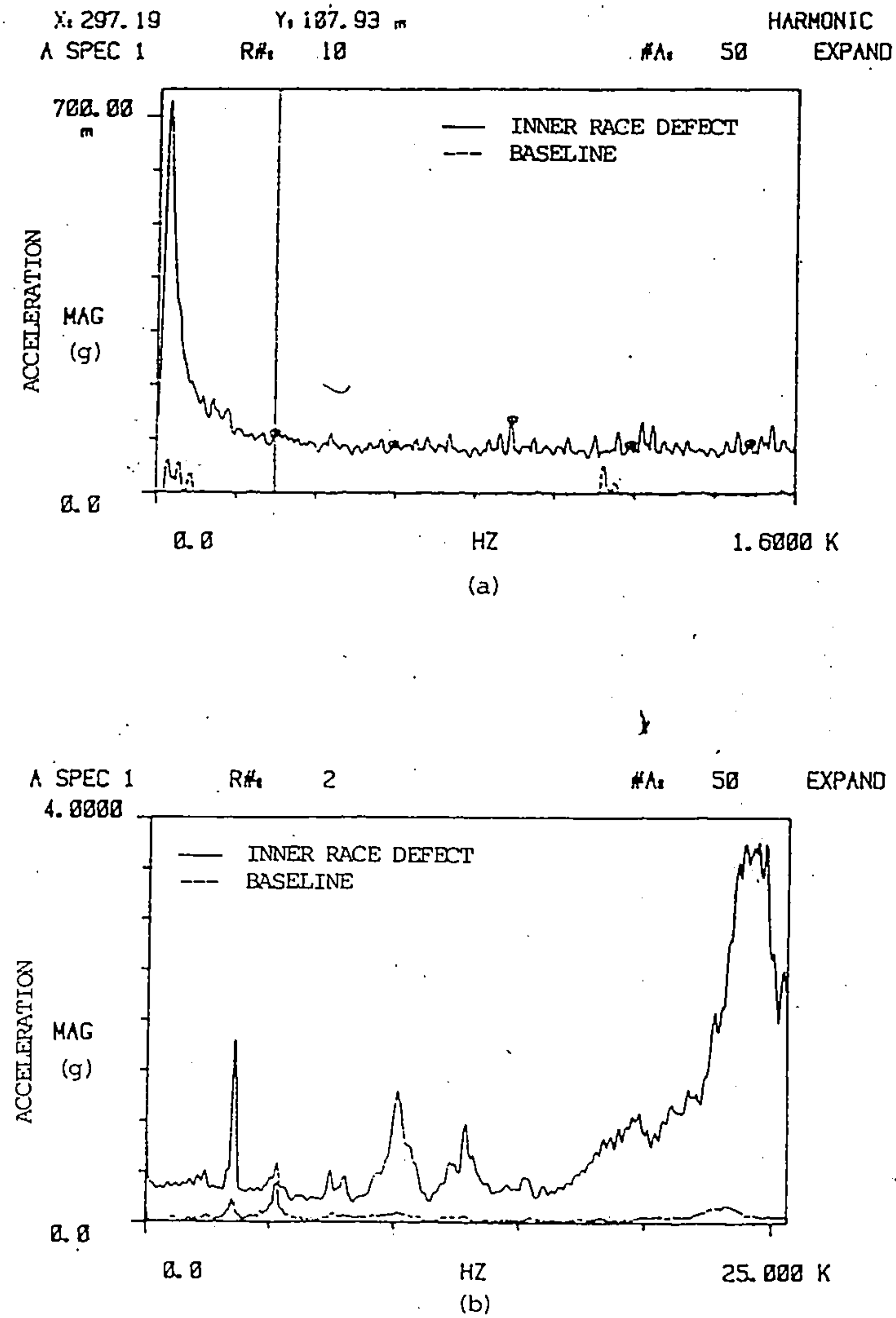
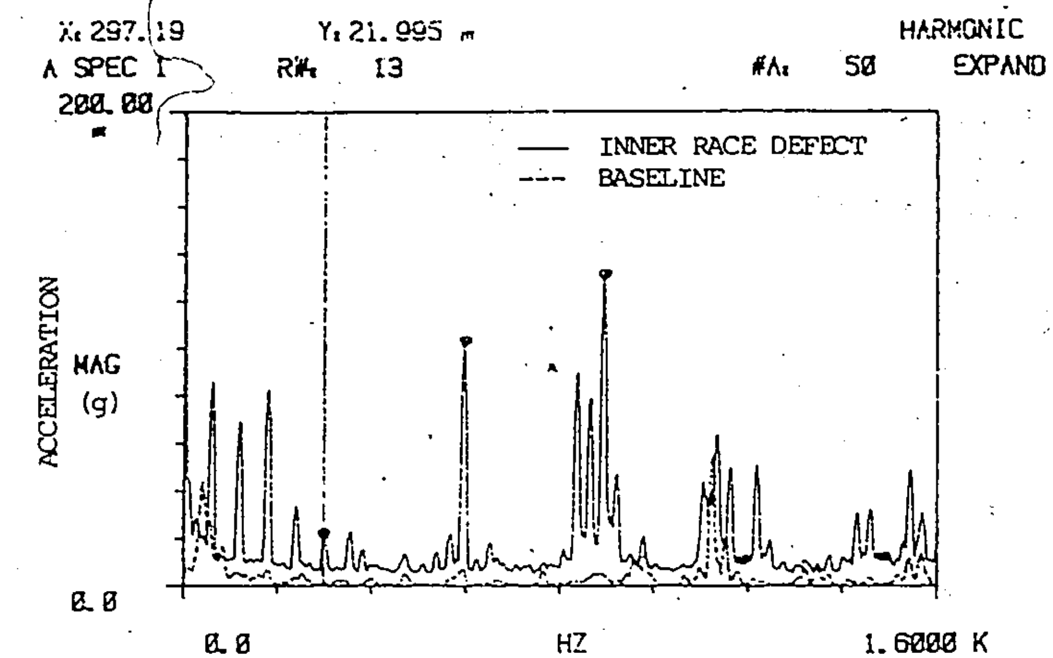
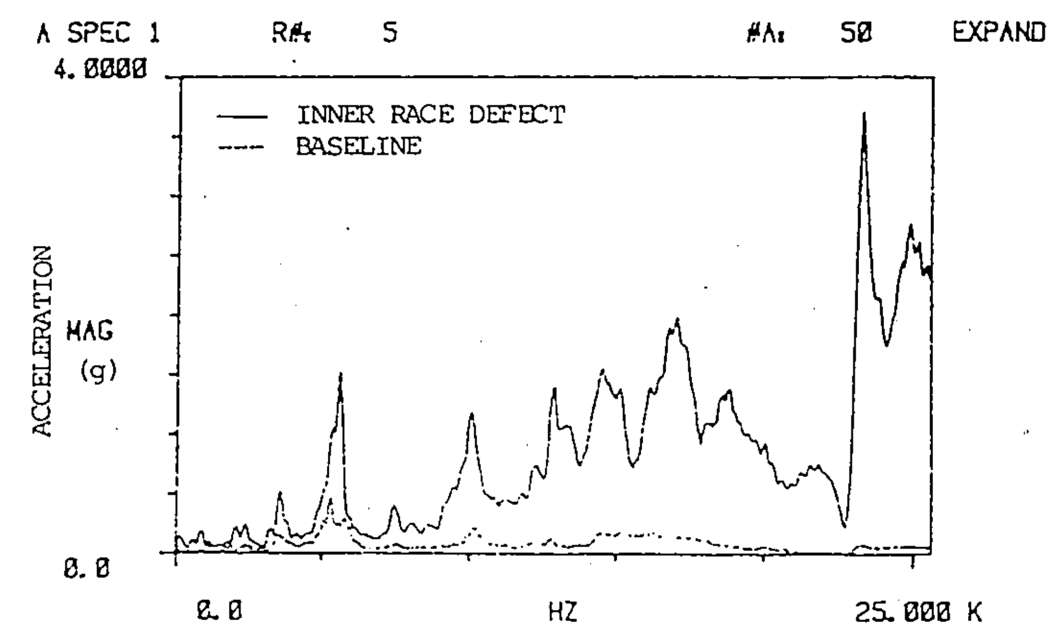


FIGURE F4: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 4P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE F5: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

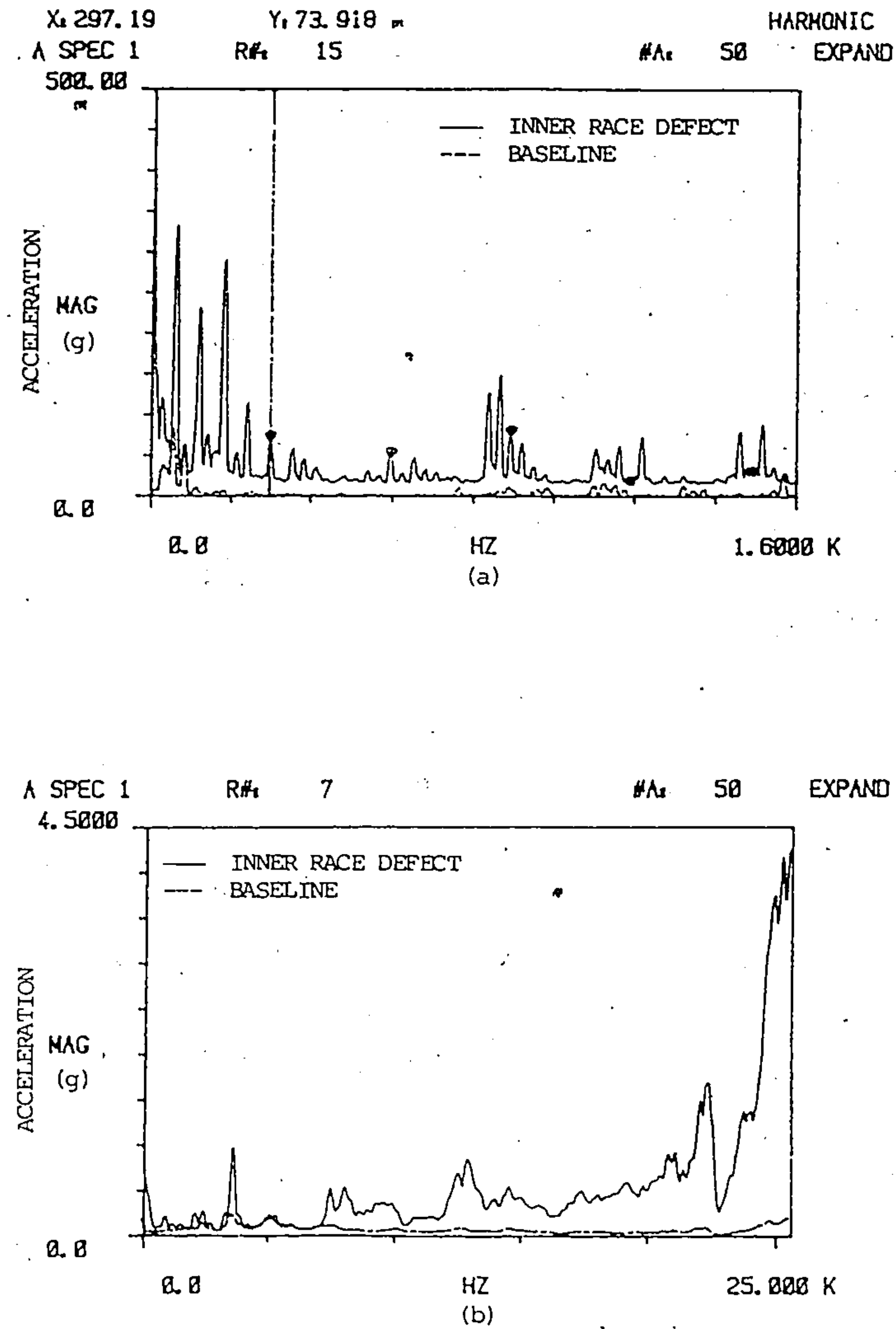


FIGURE F6: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 7B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

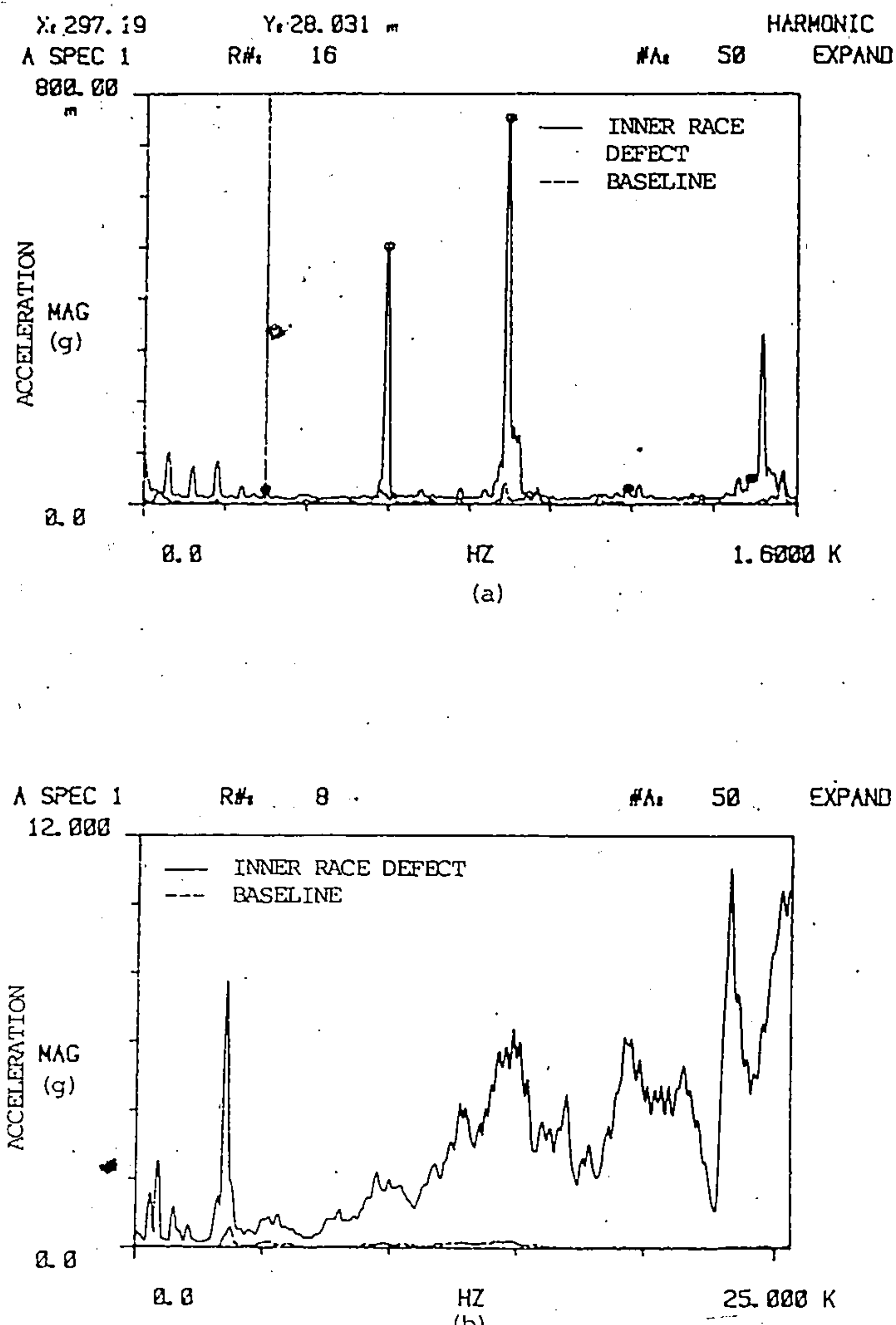
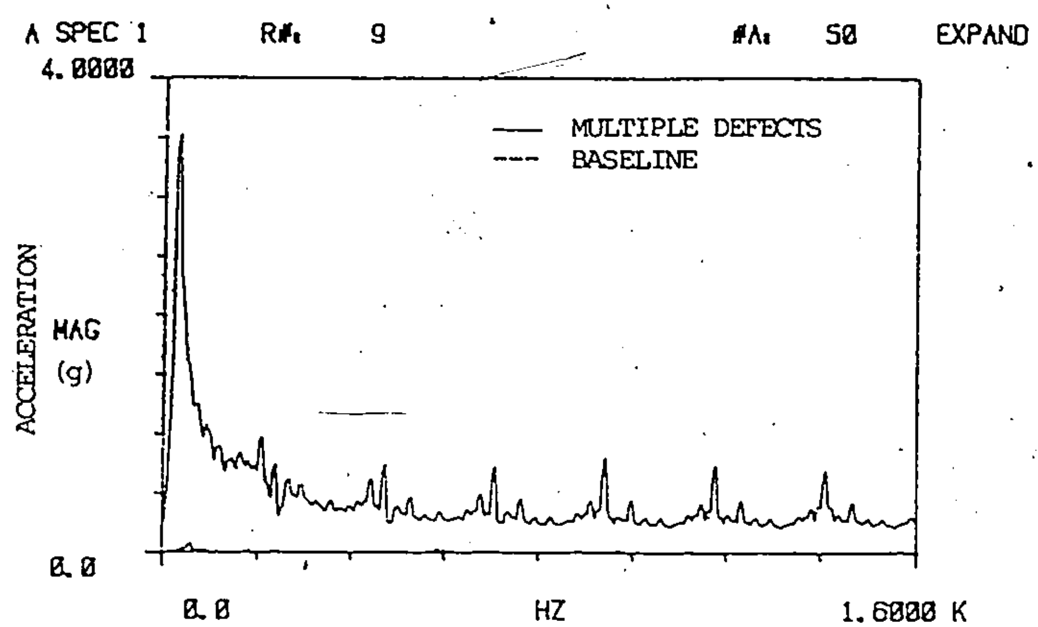


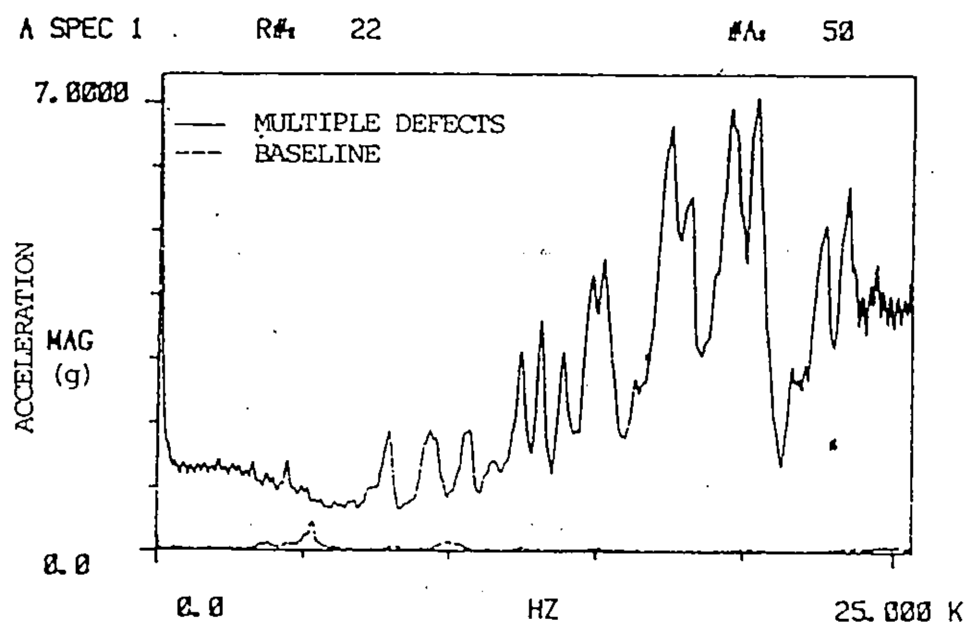
FIGURE F7: INNER RACE DEFECT SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 8B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

APPENDIX G

FREQUENCY SPECTRA FOR INDUCED MULTIPLE DEFECTS BEARING

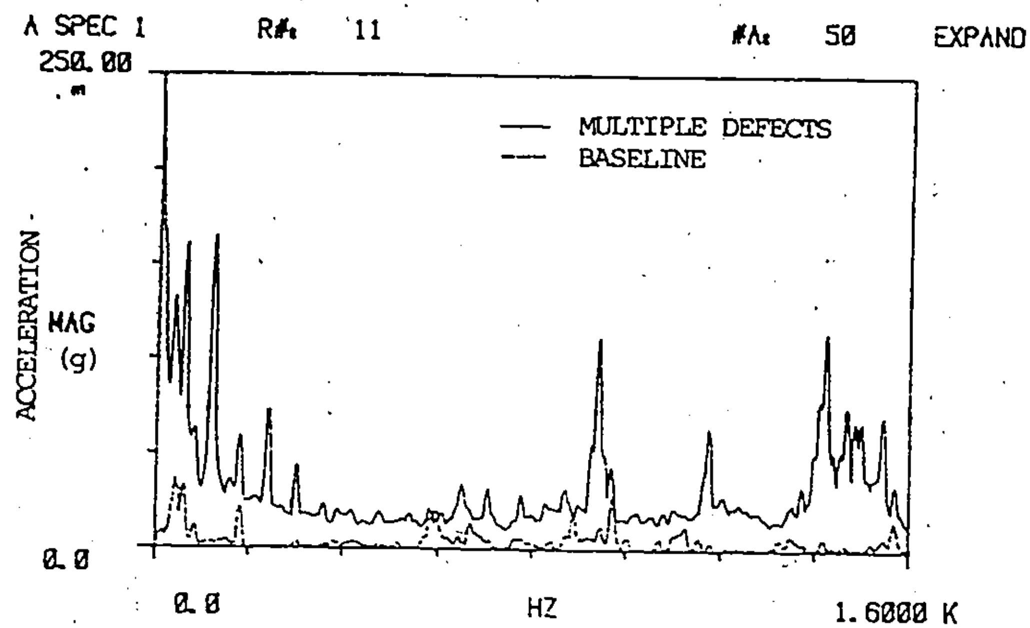


(a)

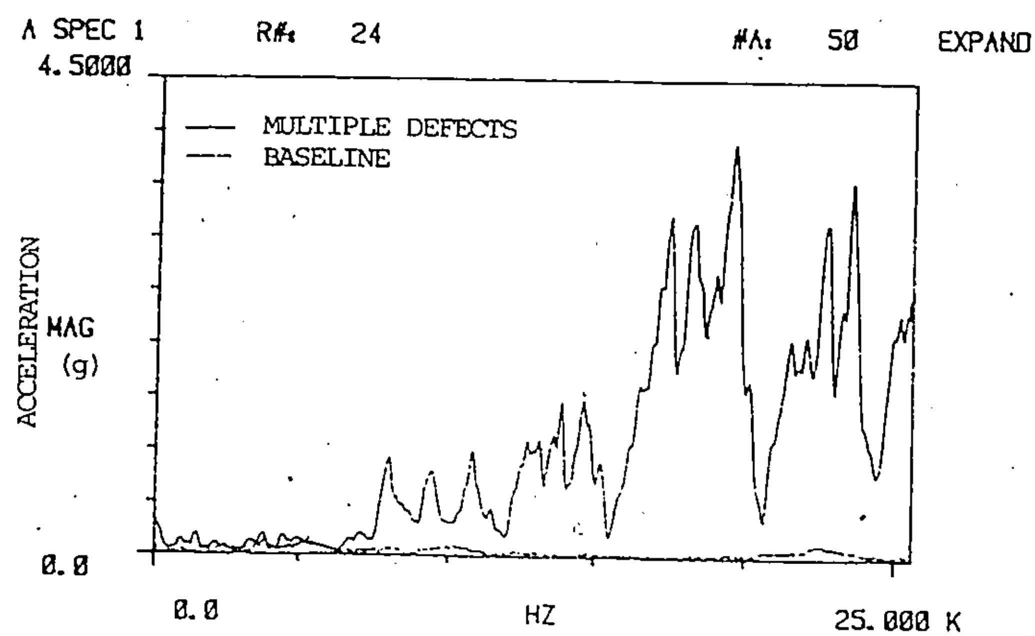


(b)

FIGURE G1: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 1P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE G2: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 2P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

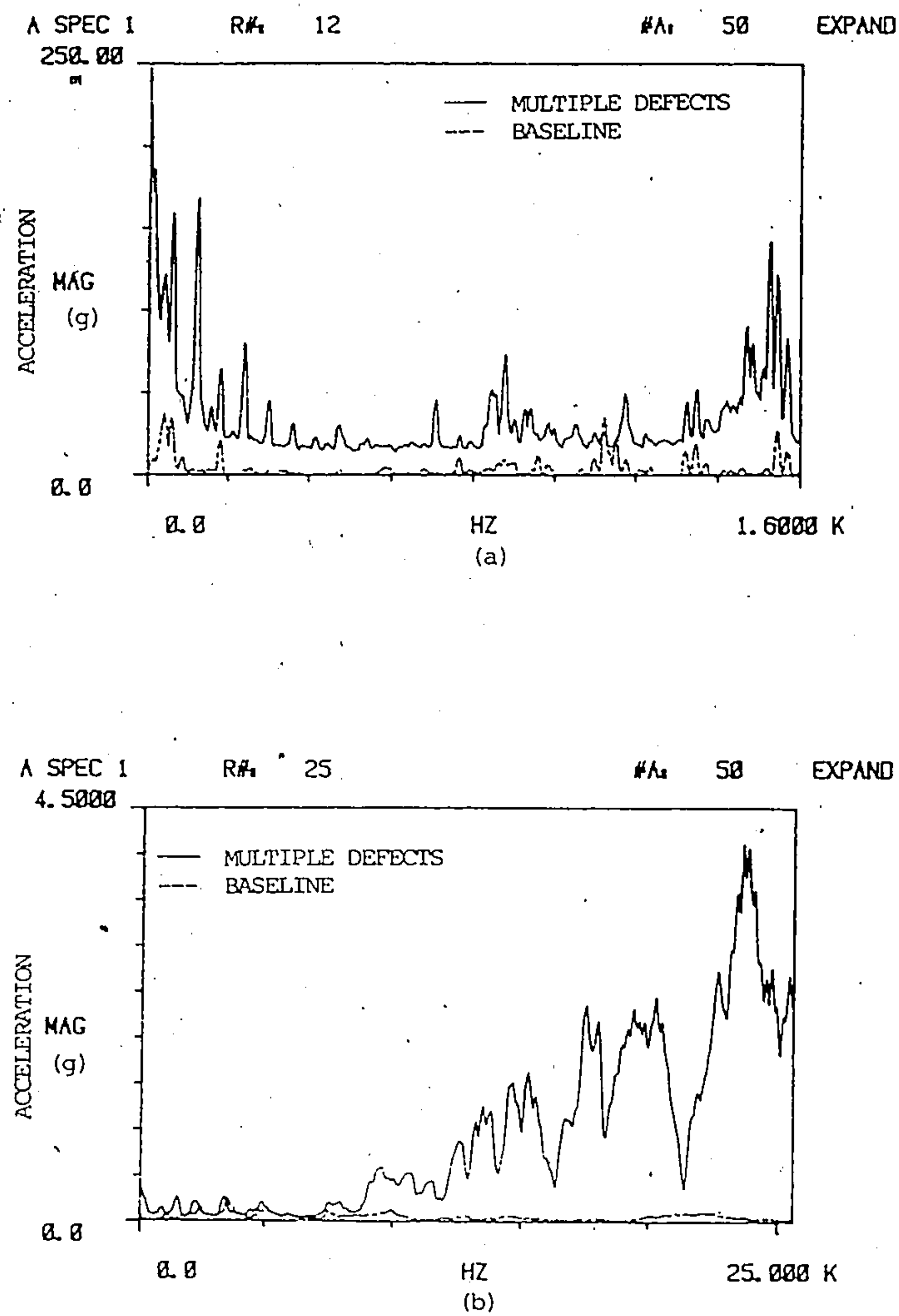
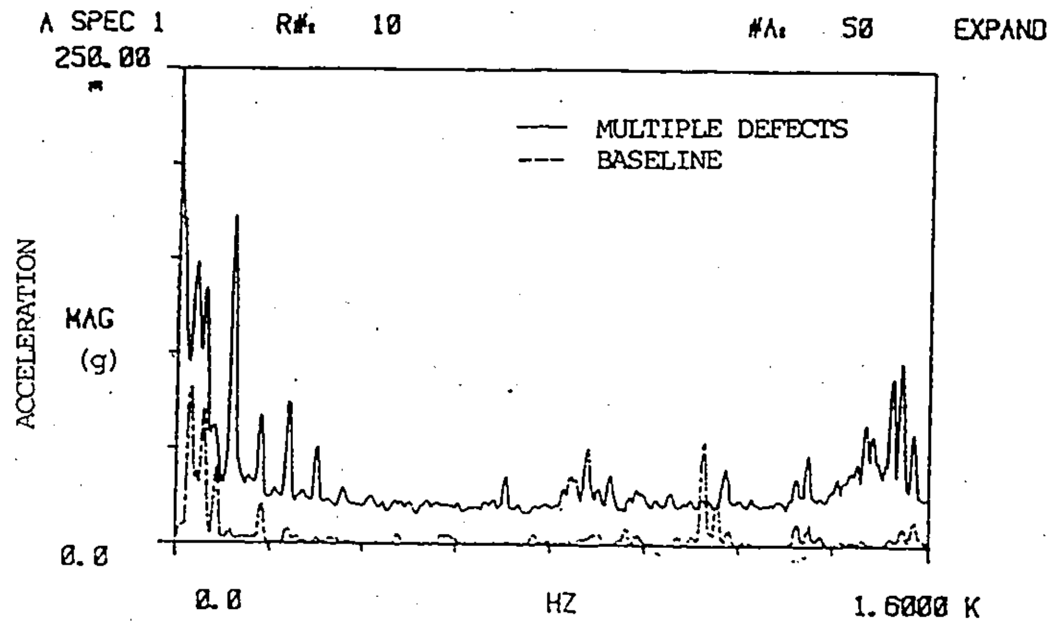
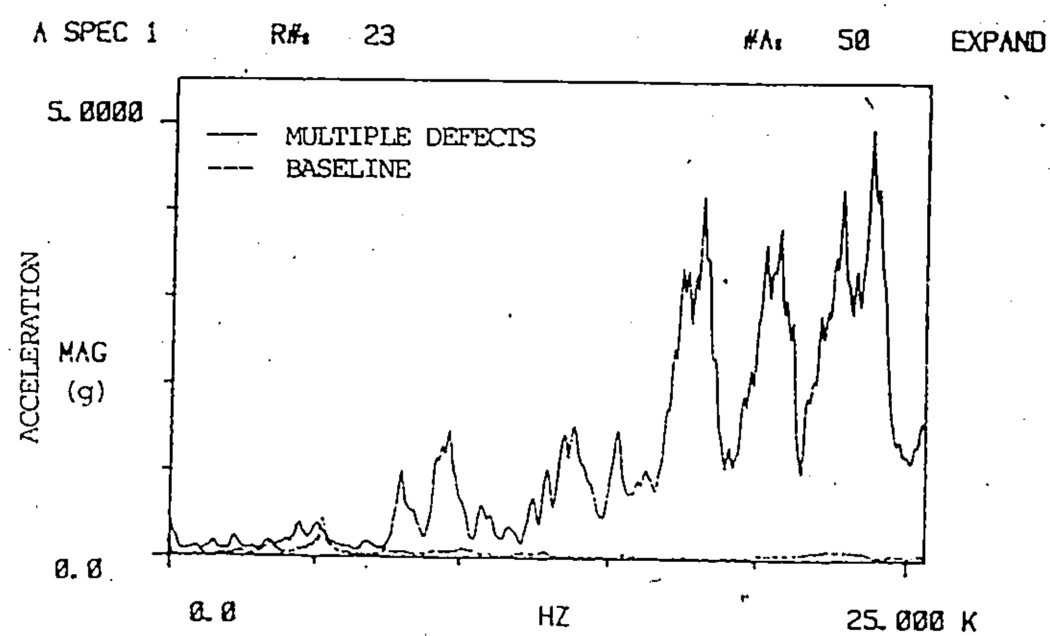


FIGURE G3: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 3P (a) 0 - 1.6kHz (b) 0 - 25.6kHz

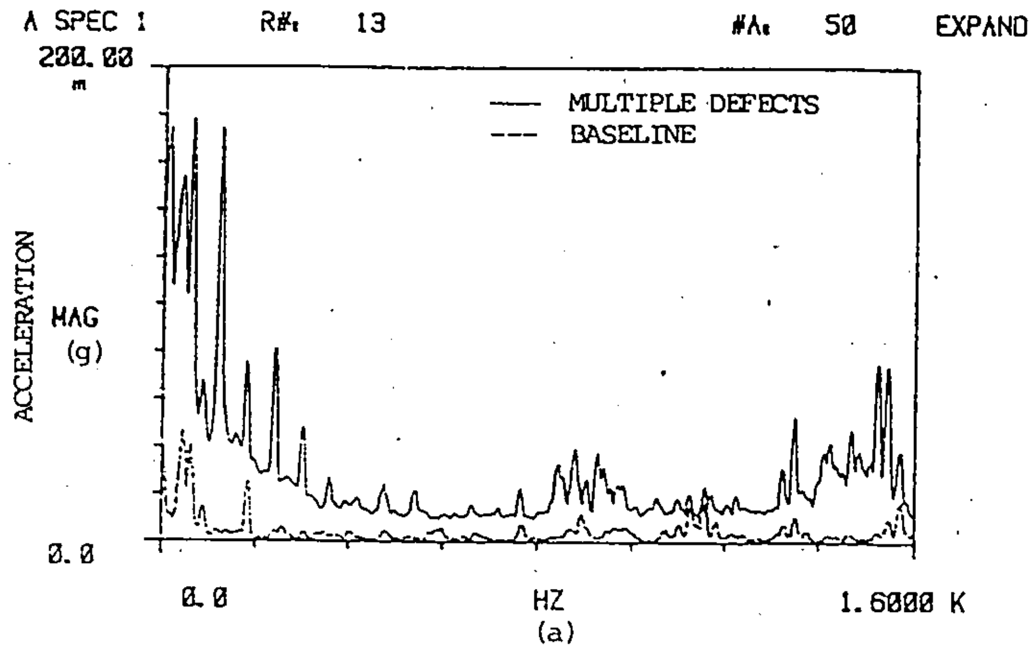


(a)



(b)

FIGURE G4: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 4P (a) 0 - 1.6kHz (b) 0 - 25.6kHz



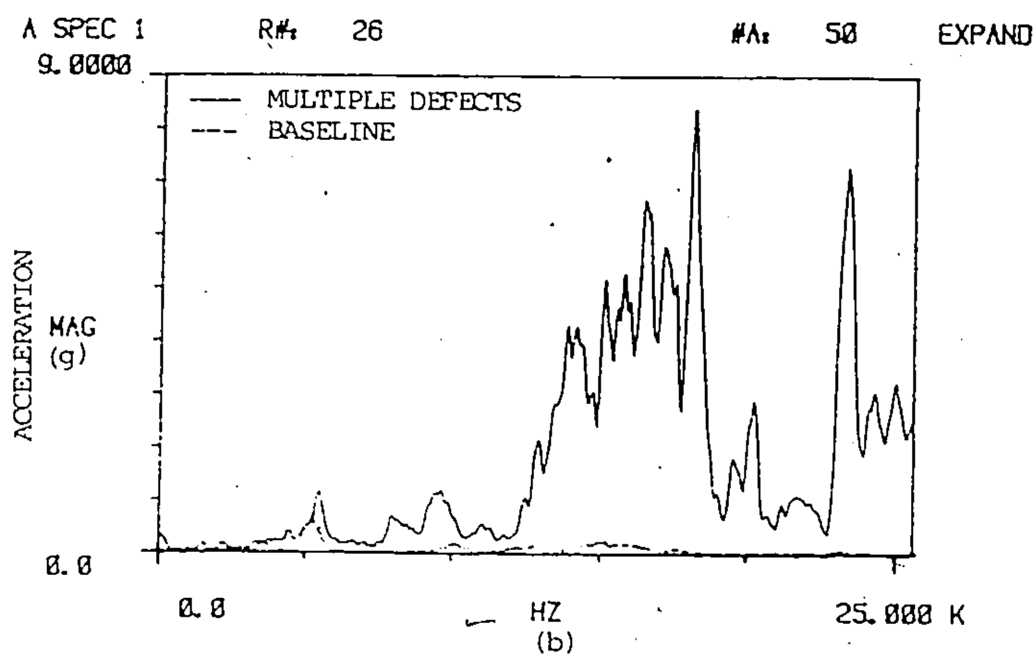
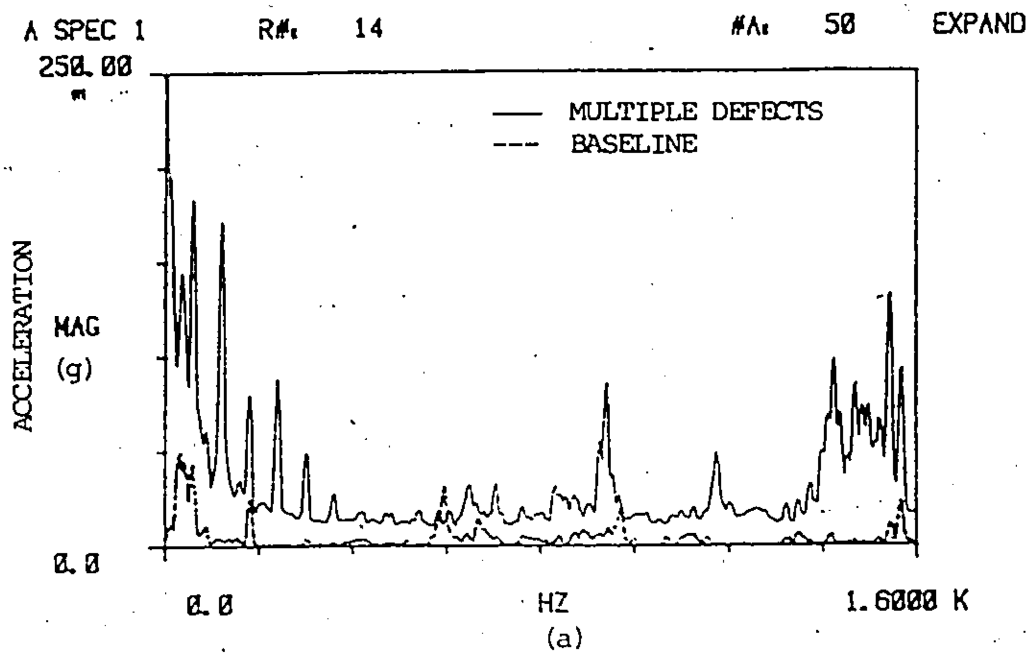


FIGURE G5: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 5B (a) 0 - 1.6kHz, (b) 0 - 25.6kHz



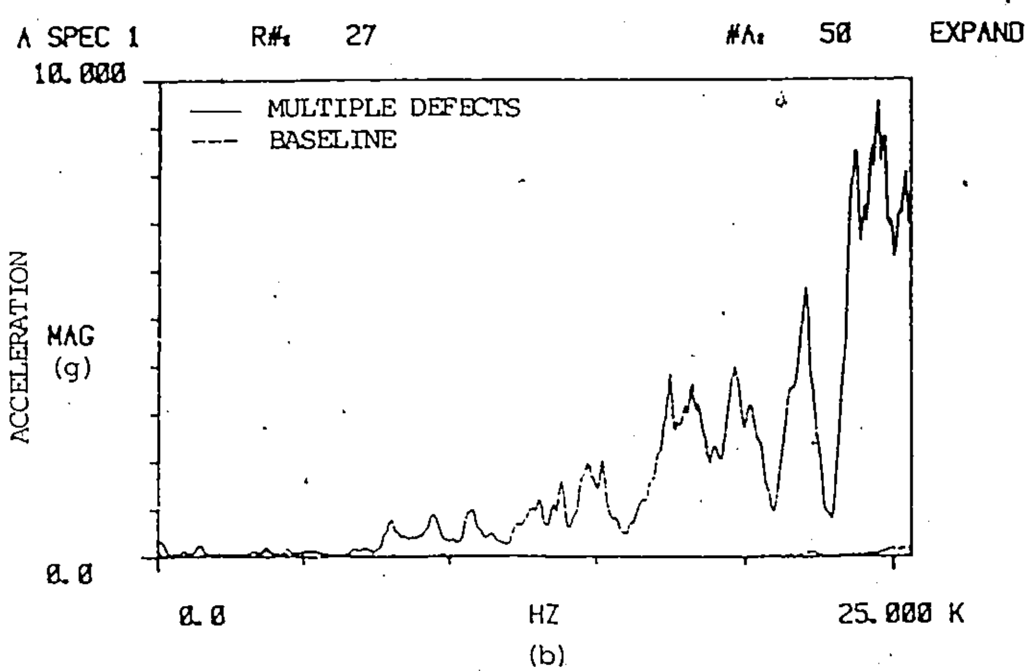
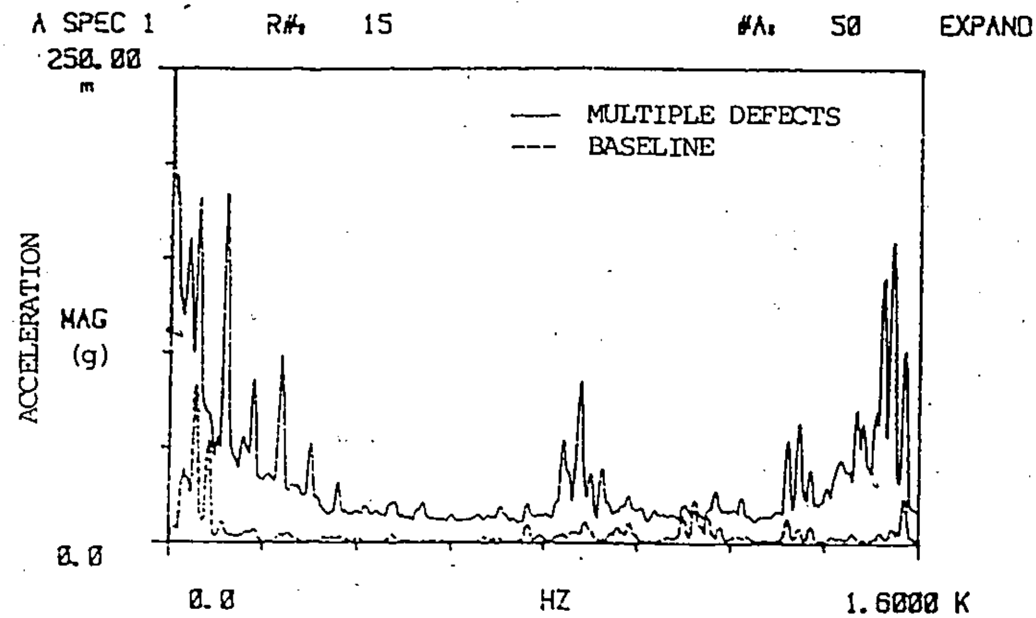
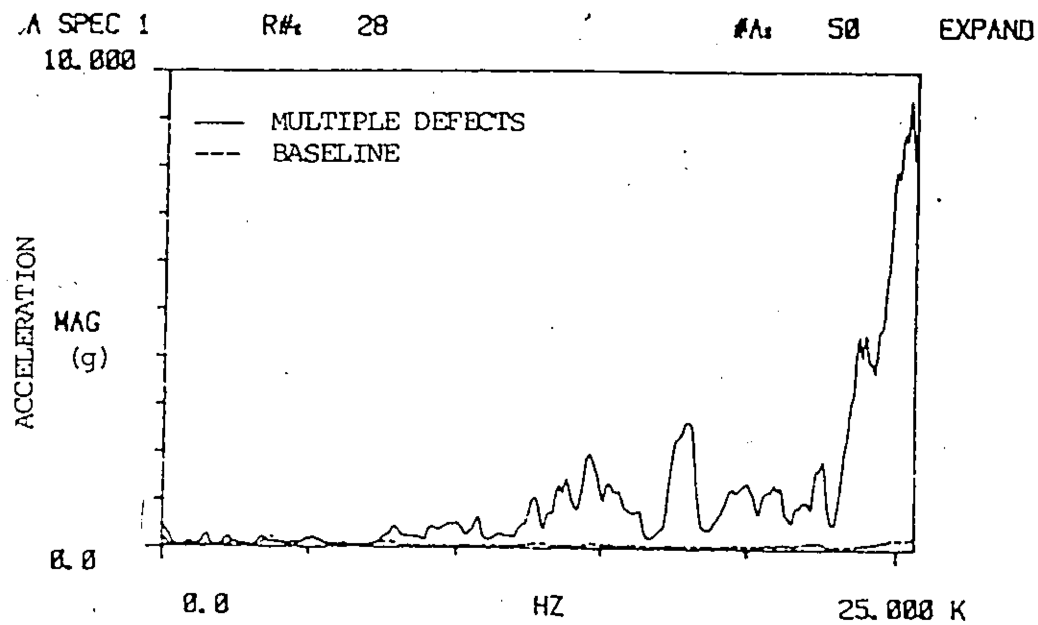


FIGURE G6: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 6B (a) 0 – 1.6kHz (b) 0 – 25.6kHz

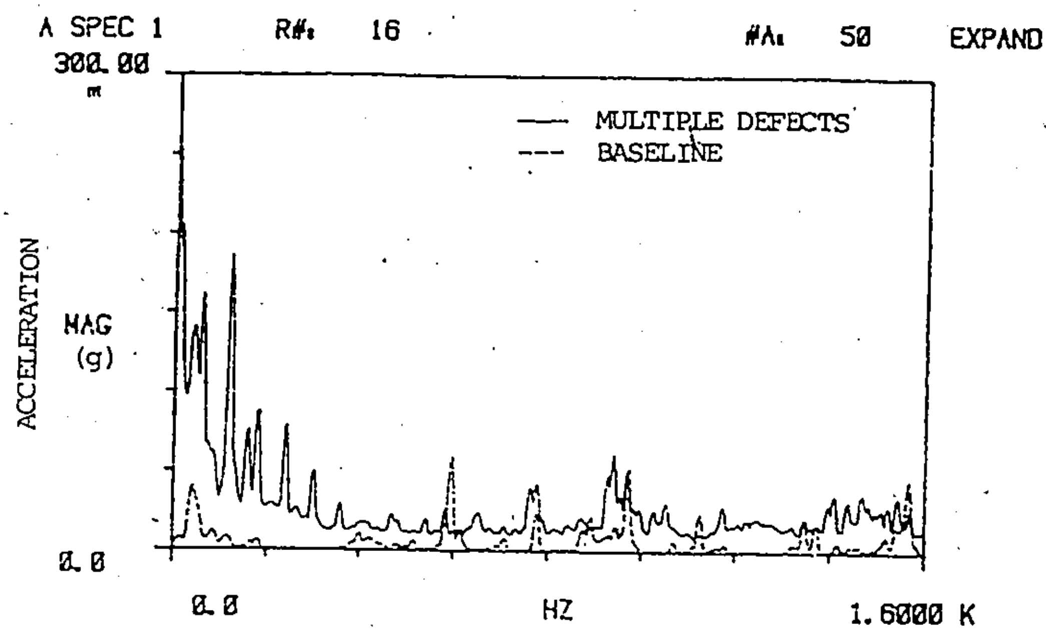


(a)

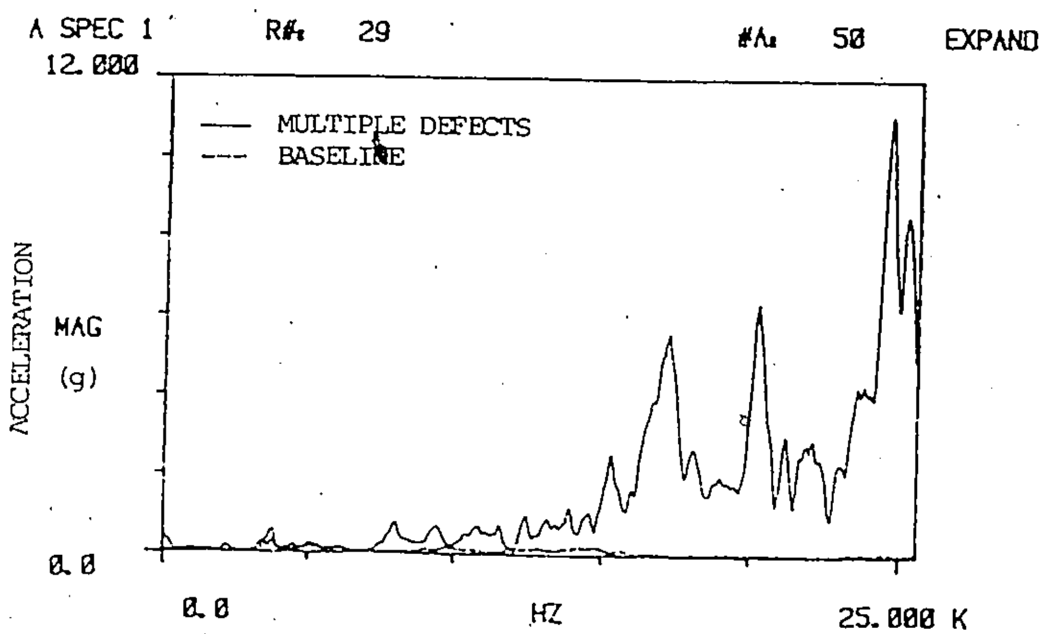


(b)

FIGURE G7: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 7B (a) 0 - 1.6kHz (b) 0 - 25.6kHz



(a)



(b)

FIGURE G8: MULTIPLE DEFECTS SPECTRUM VERSUS BASELINE SPECTRUM FOR POSITION 8B (a) 0 - 1.6kHz (b) 0 - 25.6kHz

VITA AUCTORIS

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1980	Received the Degree of Bachelor of Applied Science in Mechanical Engineering from the University of Windsor, Windsor, Ontario, Canada.
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Summer 1981	Employed as Mechanical Engineer by Timberjack Inc., Woodstock, Ontario, Canada.
1981 to Present	Presently employed as Research Engineer by F. Jos. Lamb Co. (Canada) Ltd. and a Candidate for the Degree of Master of Applied Science in Mechanical Engineering at the University of Windsor, Windsor, Ontario, Canada.